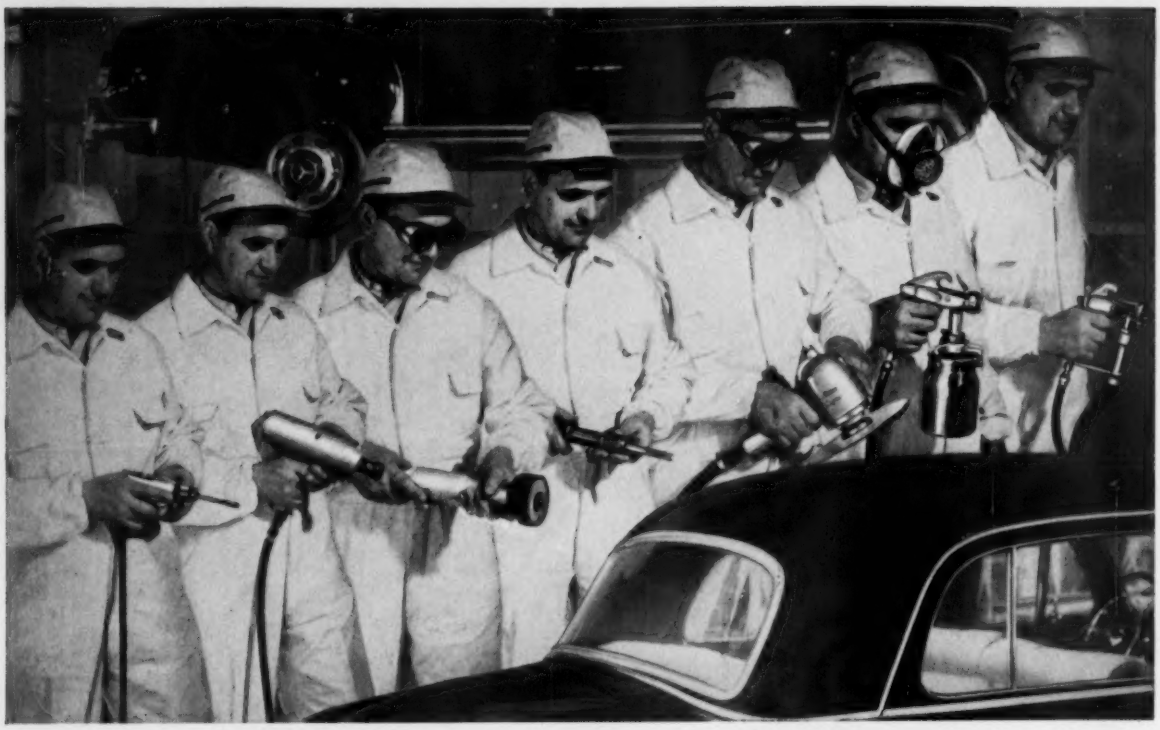
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 48 No. 10 OCTOBER 1958 PRICE: 3s. 6d.



Seven air tools from the vast range of Atlas Copco tools:—a lightweight *metal drill*; the smallest of our *impact wrenches*; a light *grinder*; a high-striking *hammer*; a heavy *grinder/polisher*; and two *spray guns*—the first specifically designed for automobile paintwork, the second for spraying compounds to coat the chassis. In each of these fields we have a series of tools to meet every need.

Atlas Copco Tools for the Motor Industry

ATLAS COPCO PNEUMATIC TOOLS SAVE TIME, MANPOWER AND COSTS

Atlas Copco pneumatic tools do most operations on assembly lines and in repair shops, and do them faster, safer and with less cost. And this is true *however small the shop.*

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(3) Pneumatic tools are safer—there's no risk of sparking, and they can be overloaded and stalled with impunity.

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Atlas Copco manufactures portable and stationary compressors, rock-drilling equipment, loaders, pneumatic tools and paint-spraying equipment. Sold and serviced by companies or agents in ninety countries throughout the world.

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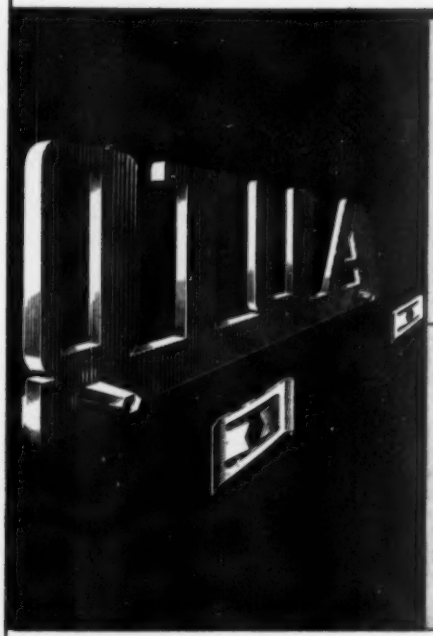

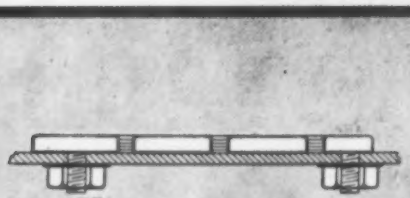

OLD WAY

Threaded studs were spot welded (or inserted if die-cast) to the nameplate. To hold this in position, washers and nuts were laboriously assembled with great difficulty especially on blind applications.

THE SALTER WAY

Studs are cast in with the nameplate, and the Fixes are simply pushed on, even on blind applications. The whole is firmly retained with an even spring tension.

save material–reduce assembly time–cut costs

When it's a question of assembling components in any engineering field, Salter Retainers are the answer. They replace nuts and bolts, screws, cotter pins, and eliminate expensive threading and machining operations. A large standard range is at your immediate disposal, and we should welcome the opportunity to assist in developing special retainers to solve your problems.

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Shell Rotella Multigrade flows freely the moment the engine starts and gives immediate engine protection.

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Because Shell Rotella Multigrade makes starting easier it reduces load on batteries.

ELIMINATES SEASONAL OIL CHANGES

Shell Rotella Multigrade has an extended viscosity range—one oil only—summer and winter.

SPECIAL BONUS FOR MIXED FLEET OPERATORS

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Shell Rotella T Multigrade 10W/30—for engines that need "Supplement 1" lubricating oils.

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SHELL

Shell Rotella Multigrade Oils

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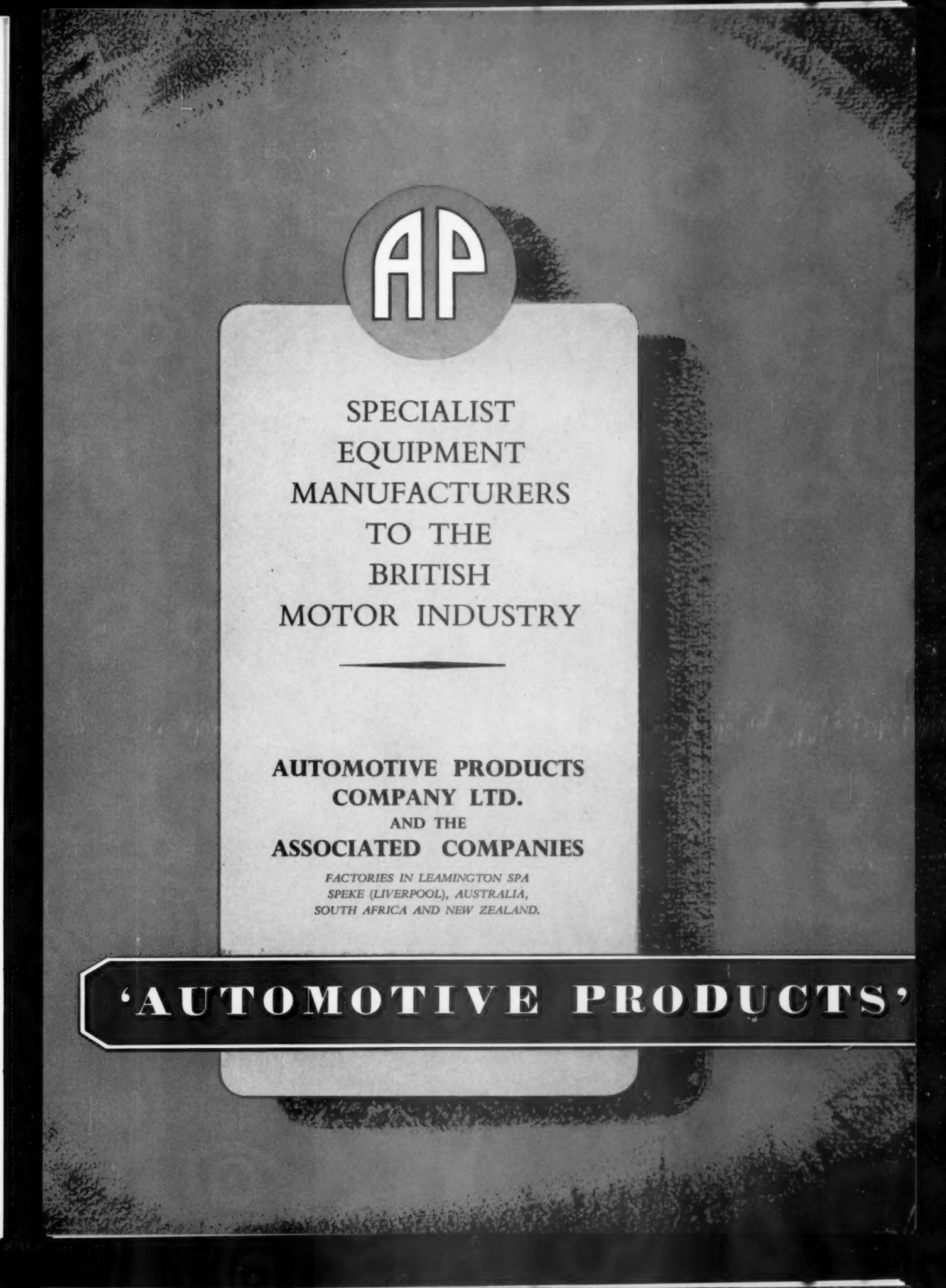
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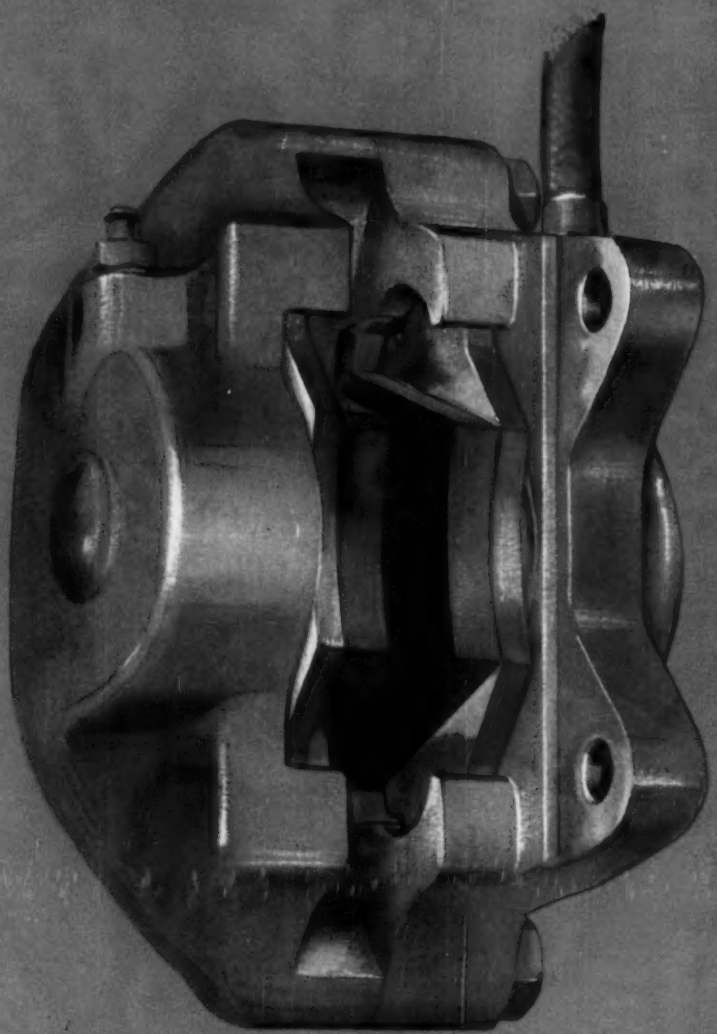
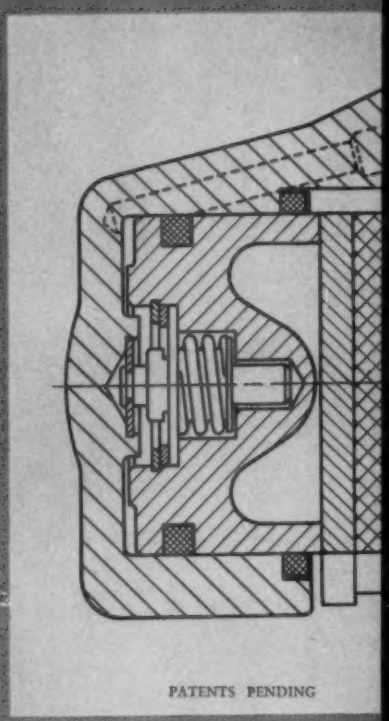
DISC





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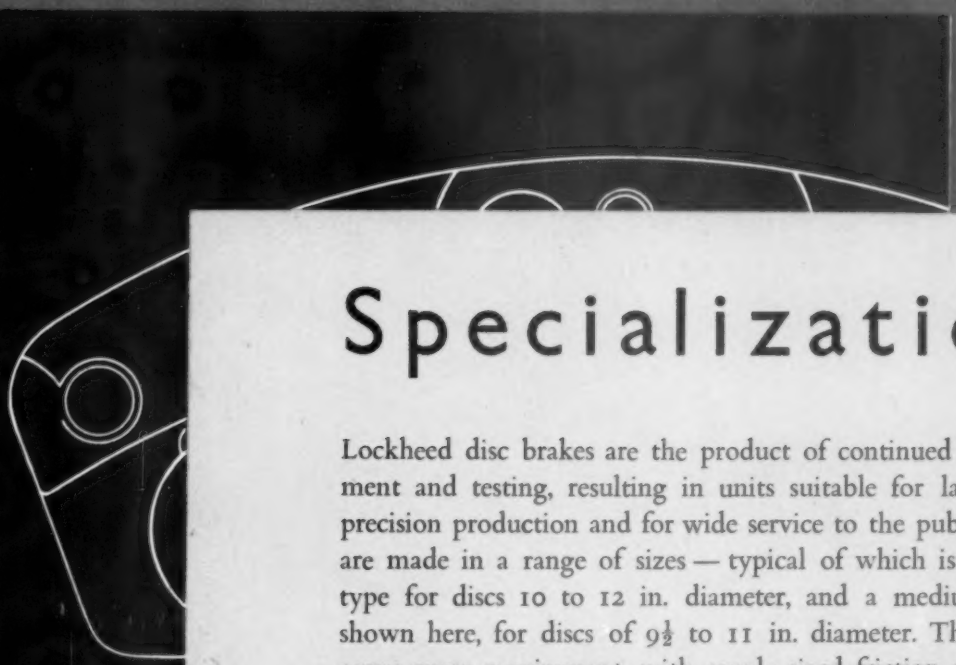
Leadership by

BRAKES



Specialization

Lockheed disc brakes are the product of continued development and testing, resulting in units suitable for large scale precision production and for wide service to the public. They are made in a range of sizes — typical of which is a heavy type for discs 10 to 12 in. diameter, and a medium type, shown here, for discs of 9½ to 11 in. diameter. These sizes cover every requirement, with ample sized friction pads that give normal mileage without replacement.

Many excellent features make these brakes outstanding:-

FREEDOM FROM FADE · AUTOMATIC ADJUSTMENT

EASE OF INSTALLATION · VERY ROBUST CONSTRUCTION

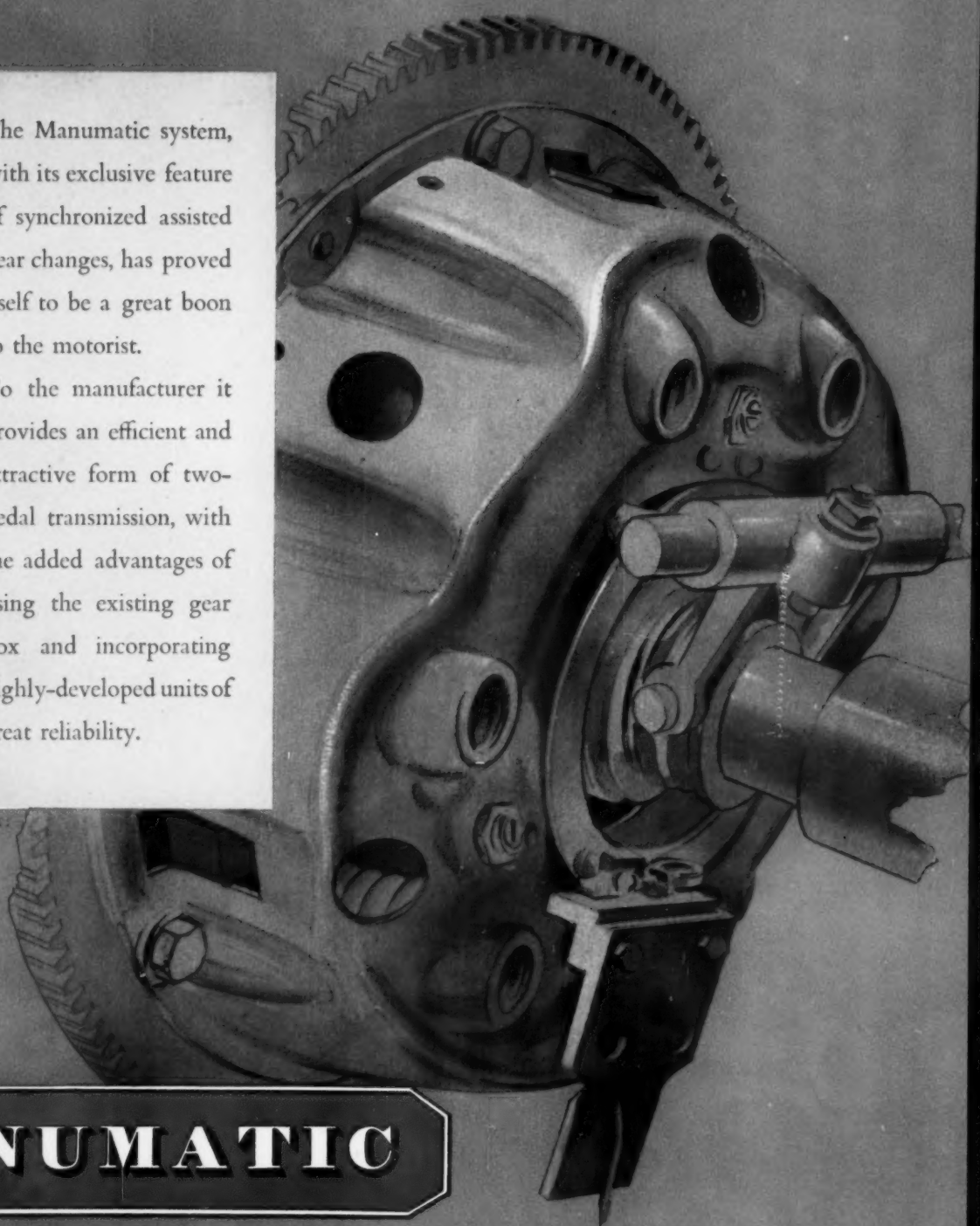
LOCKHEED

REGD. TRADE MARK: LOCKHEED

TWO-PEDAL

The Manumatic system, with its exclusive feature of synchronized assisted gear changes, has proved itself to be a great boon to the motorist.

To the manufacturer it provides an efficient and attractive form of two-pedal transmission, with the added advantages of using the existing gear box and incorporating highly-developed units of great reliability.



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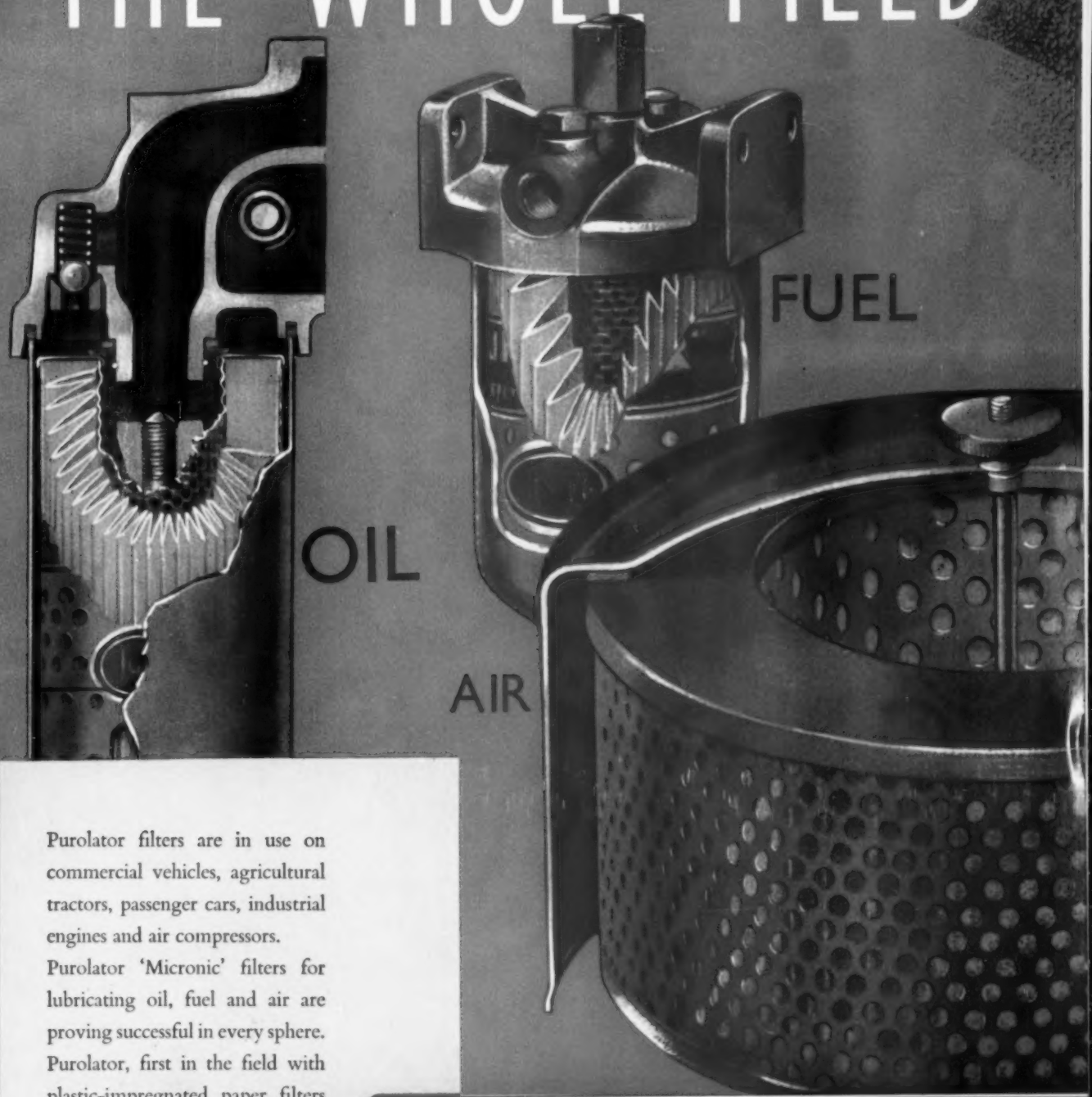
MANUMATIC

MANUMATIC

THE WHOLE FIELD



Purolator filters are in use on commercial vehicles, agricultural tractors, passenger cars, industrial engines and air compressors.

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Purolator, first in the field with plastic-impregnated paper filters for fuel and oil, now lead the way with positive air filtration.

'Micronic' air filters are available in a wide range of sizes for throughputs of up to 500 c.f.m.

PUROLATOR

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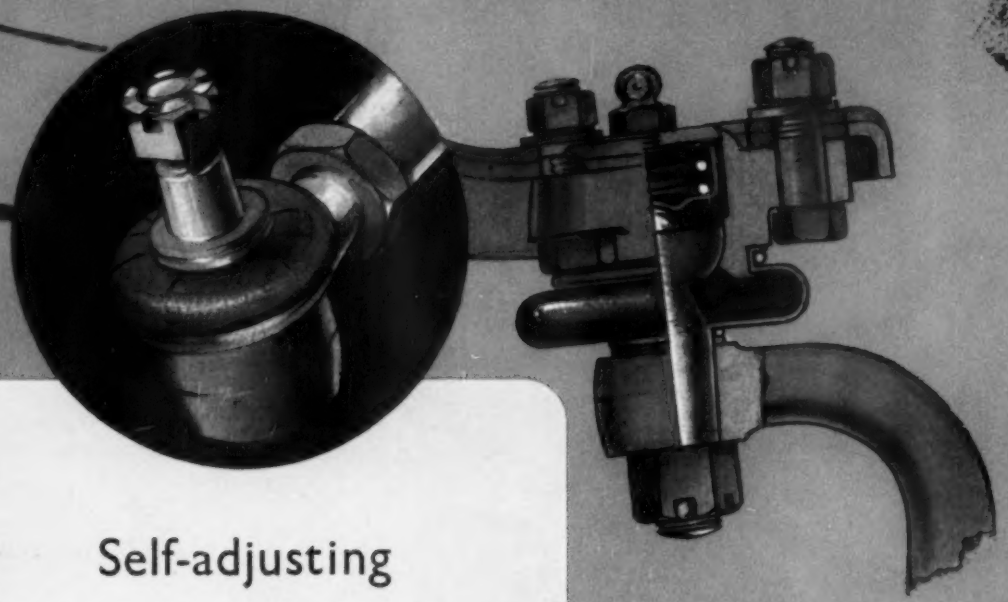
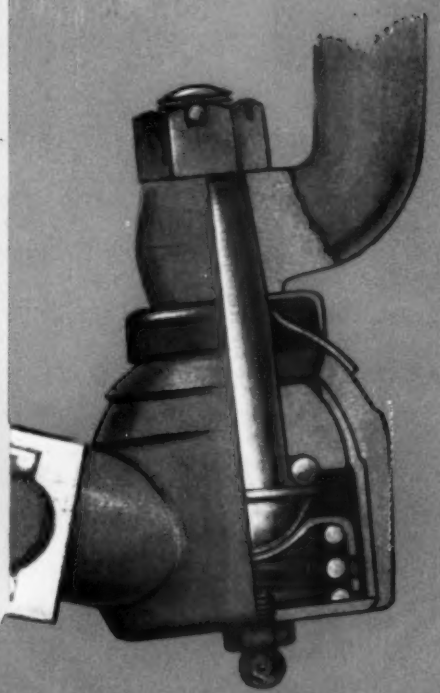


Self-adjusting

The Thompson self-adjusting ball-joint (above) eliminates dangerous backlash: friction is reduced, and construction simplified.

Thompson ball-joint suspension (at side) considerably simplifies the steering and suspension layout, facilitates assembly and overhaul and results in greater steering freedom. The upper joint is non-load-carrying and serves the double purpose of providing the second 'king-pin' bearing and the necessary articulation for the suspension linkage.
Both joints are self-adjusting.



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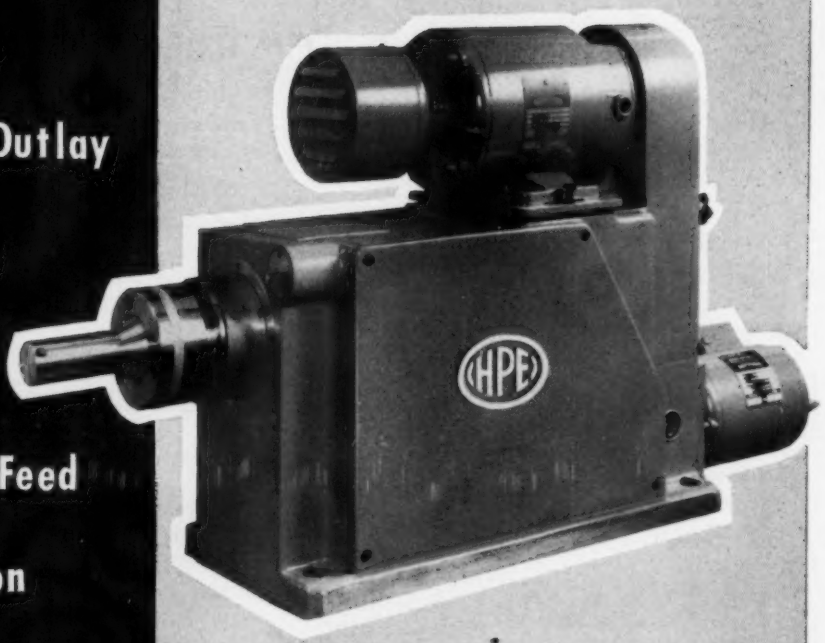

The FB4F Fine Boring Unit Head illustrated above incorporates an independent feed motion, transmitted to its spindle quill through a simple plate cam, which can be designed to give a fast approach and varying rates of feed. The robust spindle and quill, together with super precision spindle bearings, ensure a high degree of finish and roundness.

These self contained units are readily adaptable for a wide variety of applications without need for costly set ups.

Particularly suitable for mounting on simple base plates together with the necessary work holding fixture enabling single purpose units to be constructed for the minimum of time and cost. Wide range of speeds and feeds obtainable by changing feed cam and motor pulley.

"Descriptive brochure on request."

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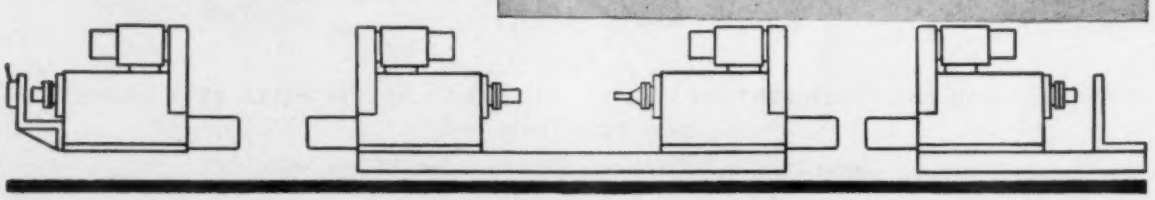
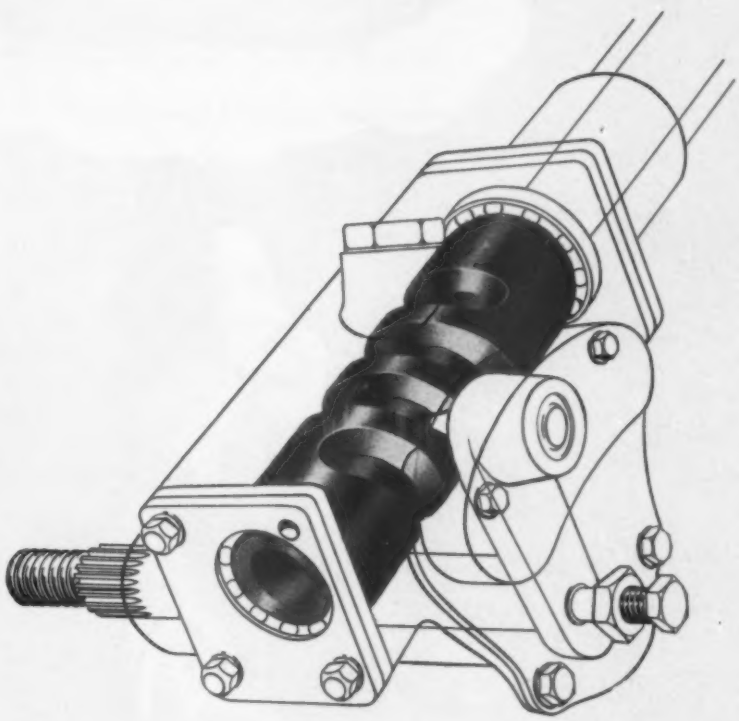


PARK GATE

QUALITY STEELS FOR AUTOMOBILE ENGINEERING



special freecutting
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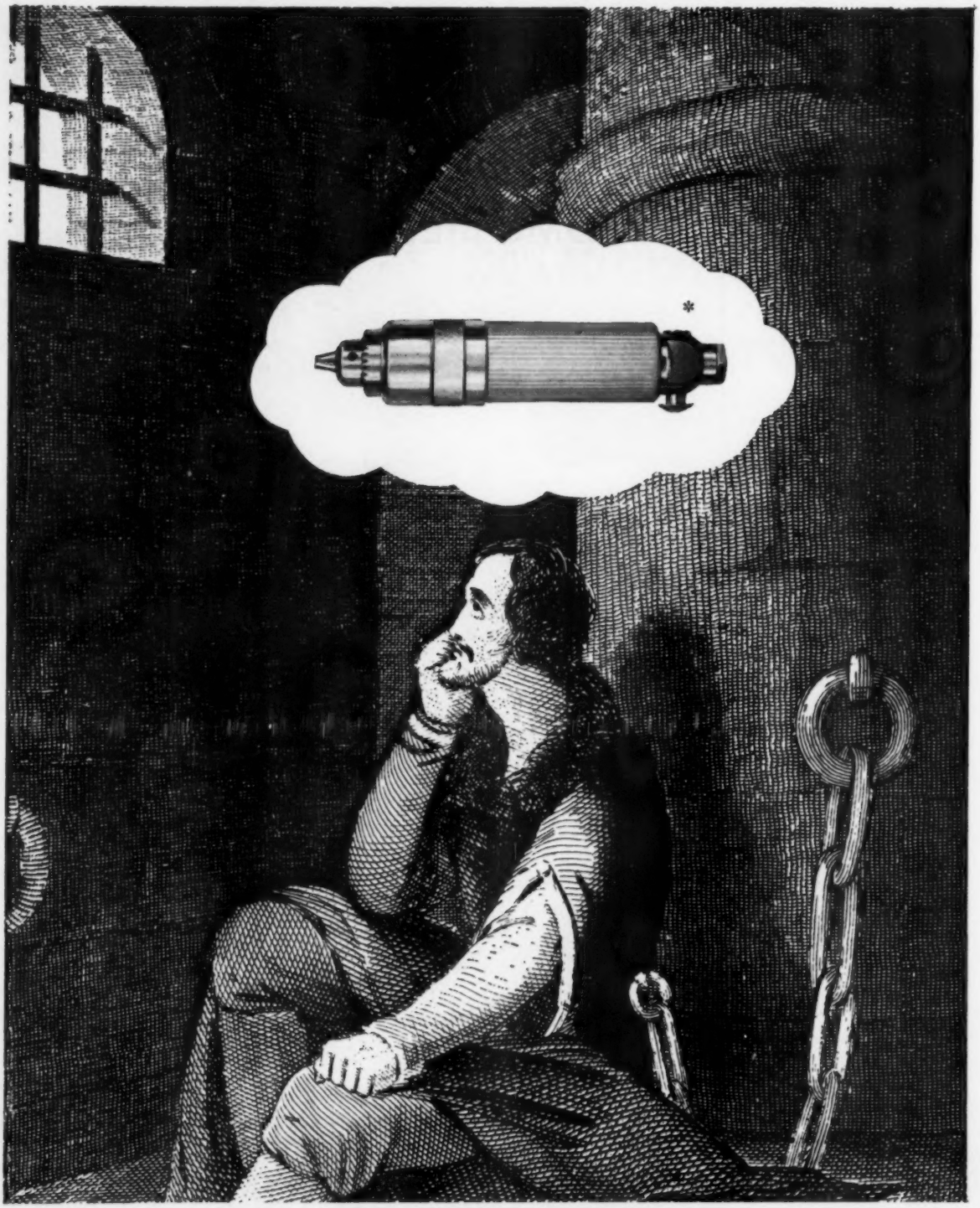

People who dream of a really quiet little drill
ought to try the ones they're now turning out at Hendon.

* *This dream appears by kind permission of Desoutter Brothers Limited, makers of the new silent Mighty Atom and Power Atom pneumatic drills.*

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2 minutes and 1 second

This nickel chrome molybdinum steel gearbox mainshaft, nearly 16″ long was produced from a forging in two minutes and one second on a Churchill-Redman P.5. Lathe. In this time 3½ lbs. of metal were removed.

This is the usual high speed of operation obtainable on a **Churchill-Redman. P.5. Multi-tool and Profiling Machine.** Ample power and the speed of feed make substantial new machining economies possible.

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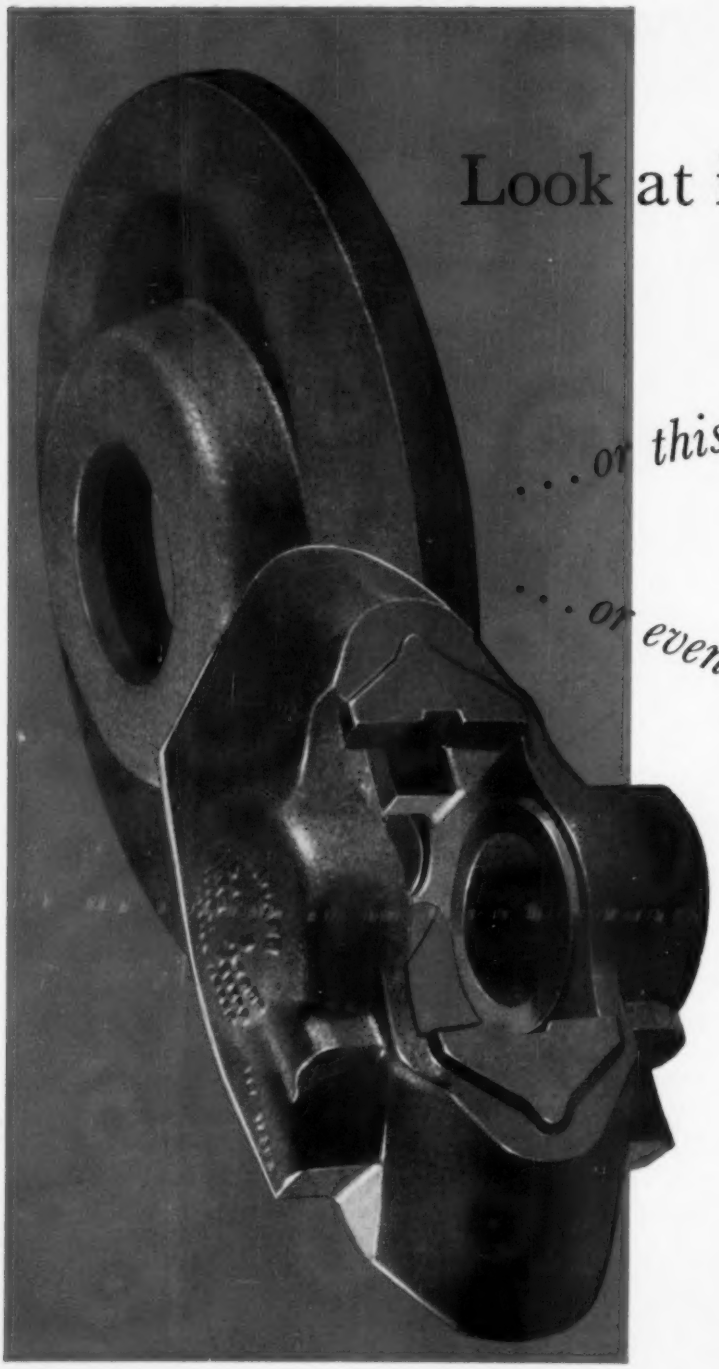

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. . . or this way . .

. . . or even this way

. . . the best brake discs
and calipers are in high duty
cast iron by

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Speed of threading bolts, studs and rods.

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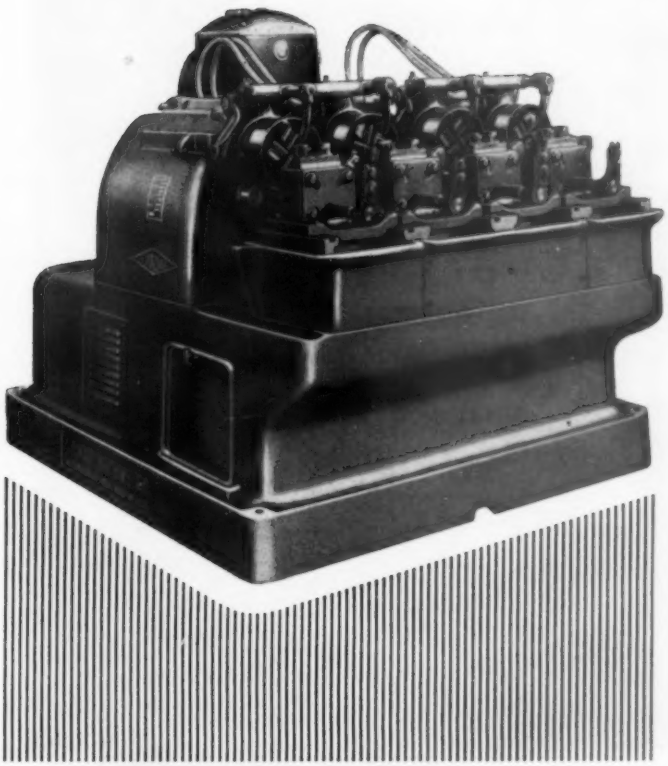


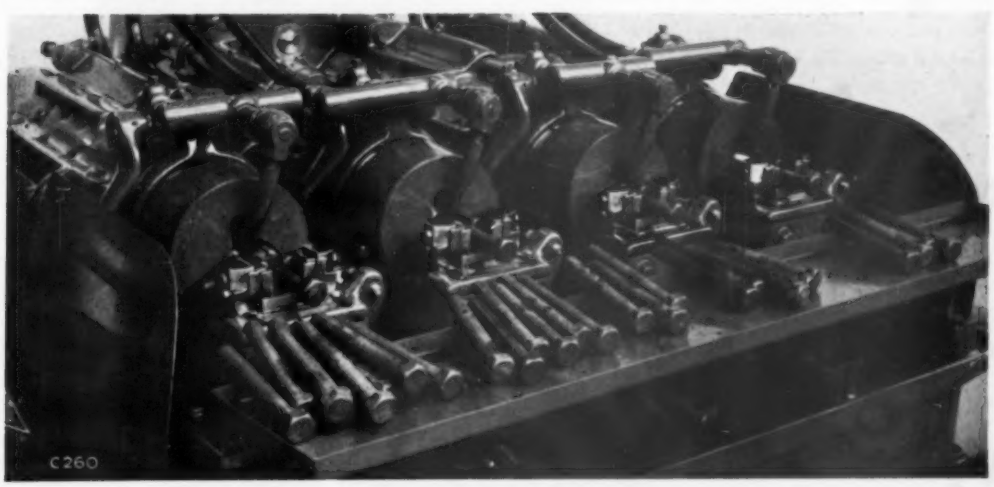
For the high production threading of bolts, studs and rods. Two sizes for threads from $\frac{1}{4}$" to $\frac{3}{4}$" × 3$\frac{1}{2}$" long and $\frac{1}{2}$" to 1" × 3$\frac{1}{2}$" long.

The operator simply loads and unloads the grips as they automatically open. Production ranges from 500 to 2,000 per hour, depending on diameter and thread length. Standard or special gripping mechanism fitted.

The slow spindle speed feature of this machine increases the life of the chasers.

The illustration below shows :

Cutting $\frac{3}{4}$" diameter by 1$\frac{1}{2}$" long Whit. threads on black bolts. Production, 1,650 bolts per hour. Rate of production largely depends on the speed of the operator loading and unloading the grips.



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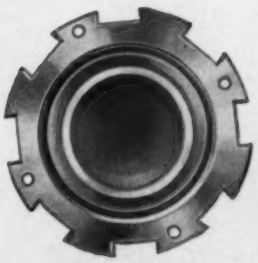
* *Polytetrafluoroethylene*



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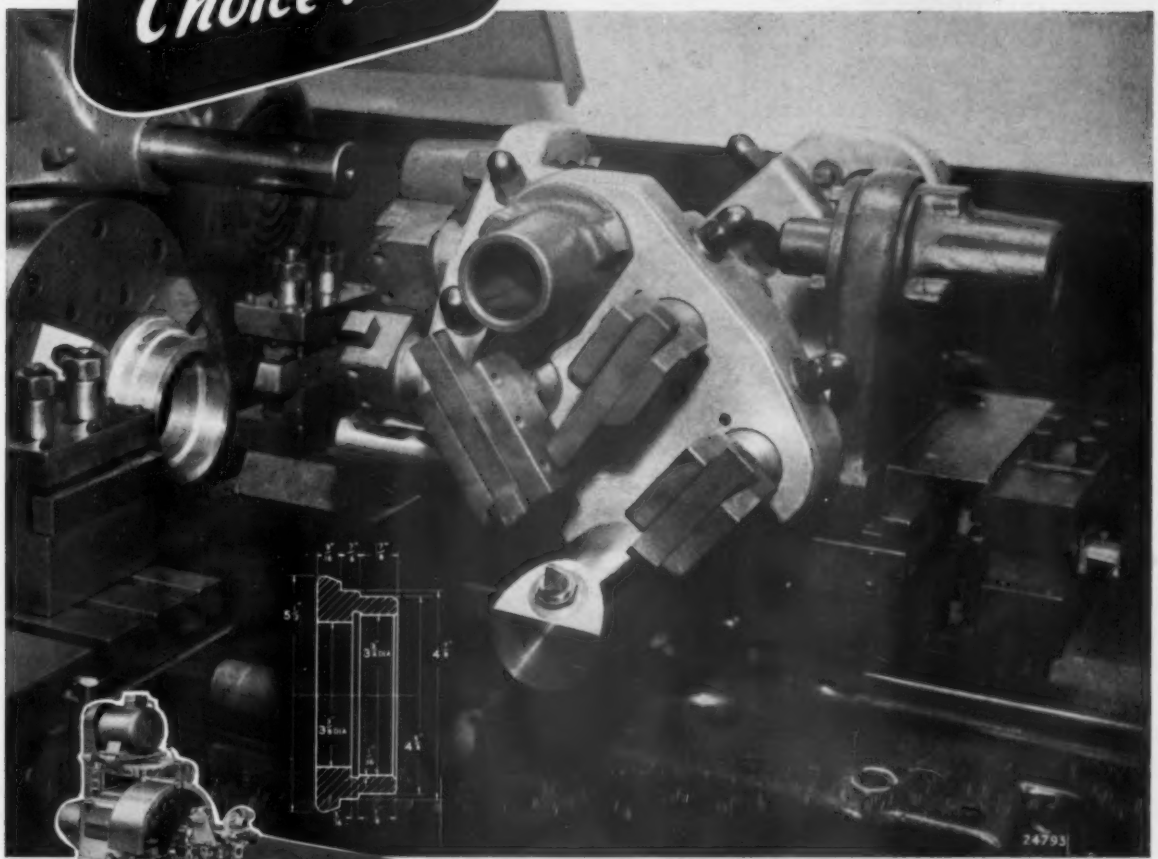

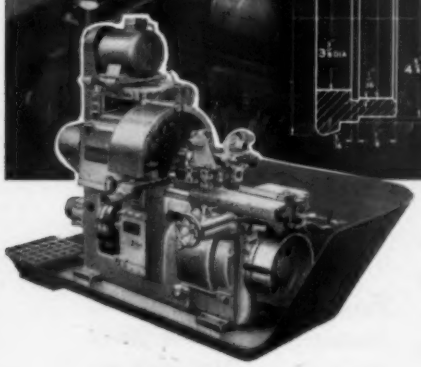

Tooling arrangement on a Herbert No. 3A Auto lathe for the production of thrust plates from 0·45% carbon-steel forgings in a total time of 3·16 minutes each.

For high-production repetition work with consistent accuracy.
Greatest output per foot of floor space at lowest labour cost per piece—one operator and one tool setter can keep from four to six machines in continual operation.
Power, rigidity and speed ranges to take full advantage of Ardoloy and other carbide tooling.
All operations except chucking and removing the work are entirely automatic.

Herbert Auto-lathes are made in five sizes: Auto Junior—8½″ swing, No. 3A.—12½″ swing, No. 4—16½″ swing, No. 5A.—25″ swing and No. 5A/33—33½″ swing.

Available for Early Delivery

Ask for brochure "Production on Herbert Auto-lathes."

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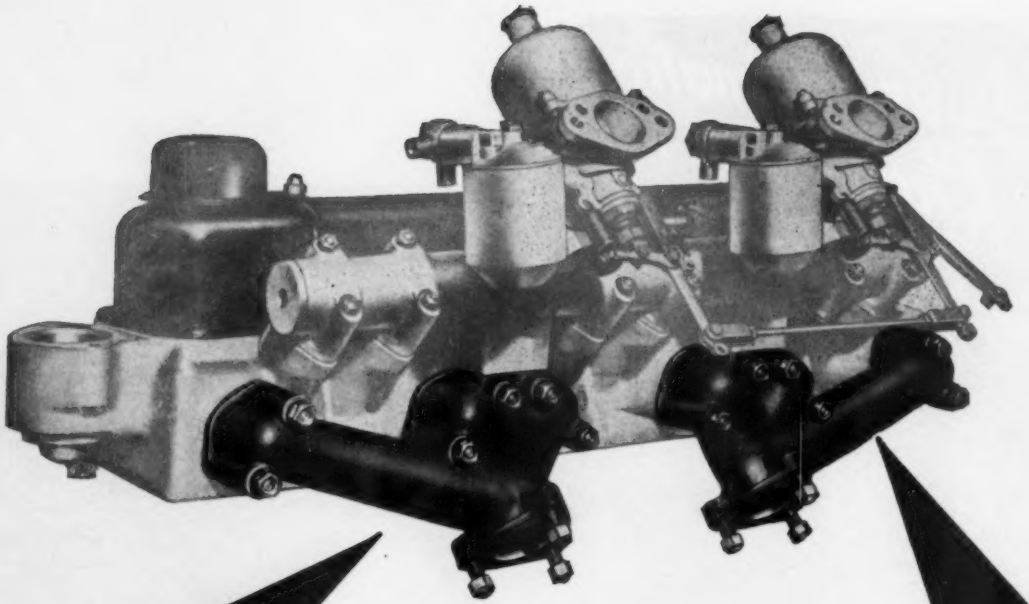

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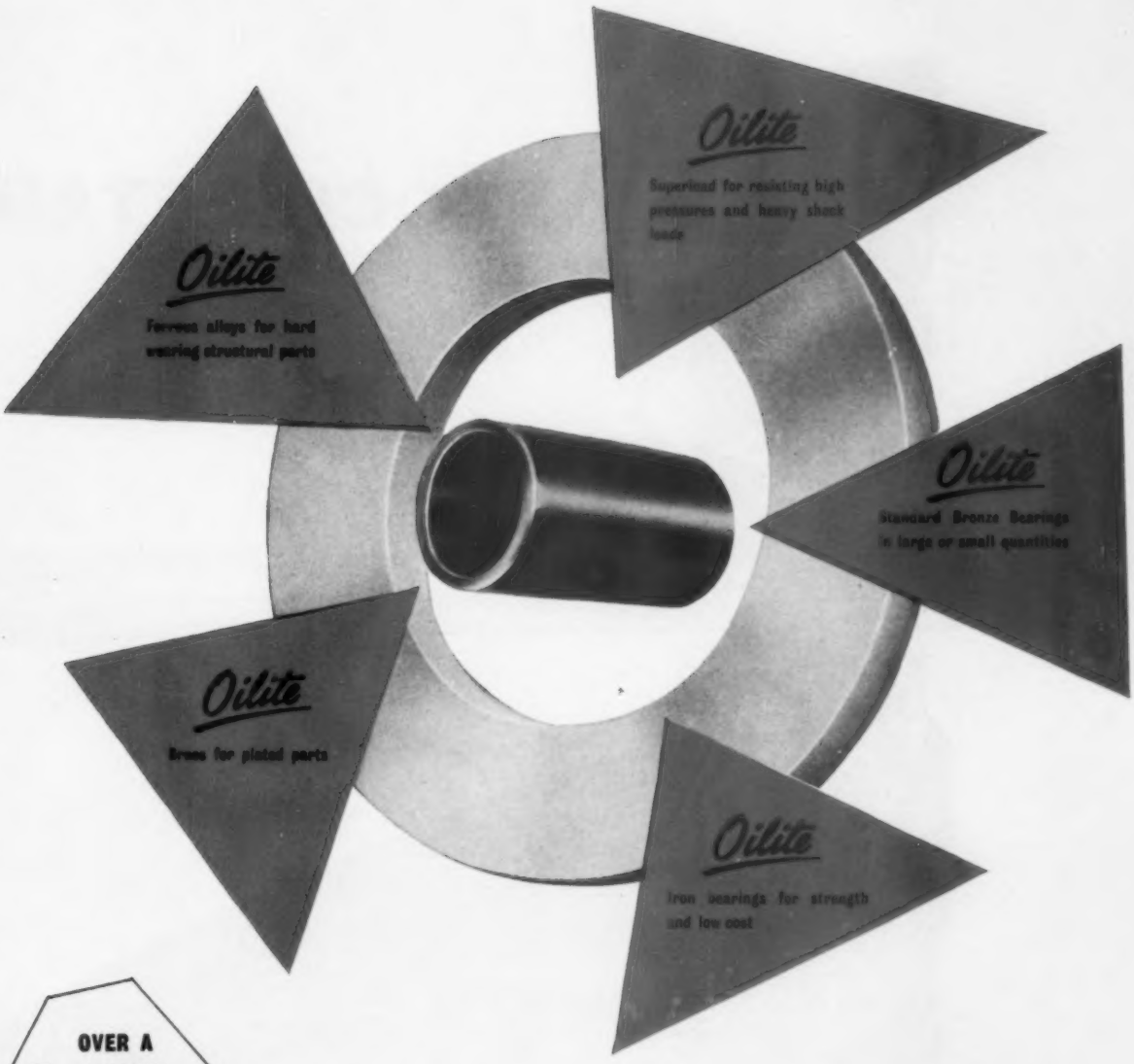
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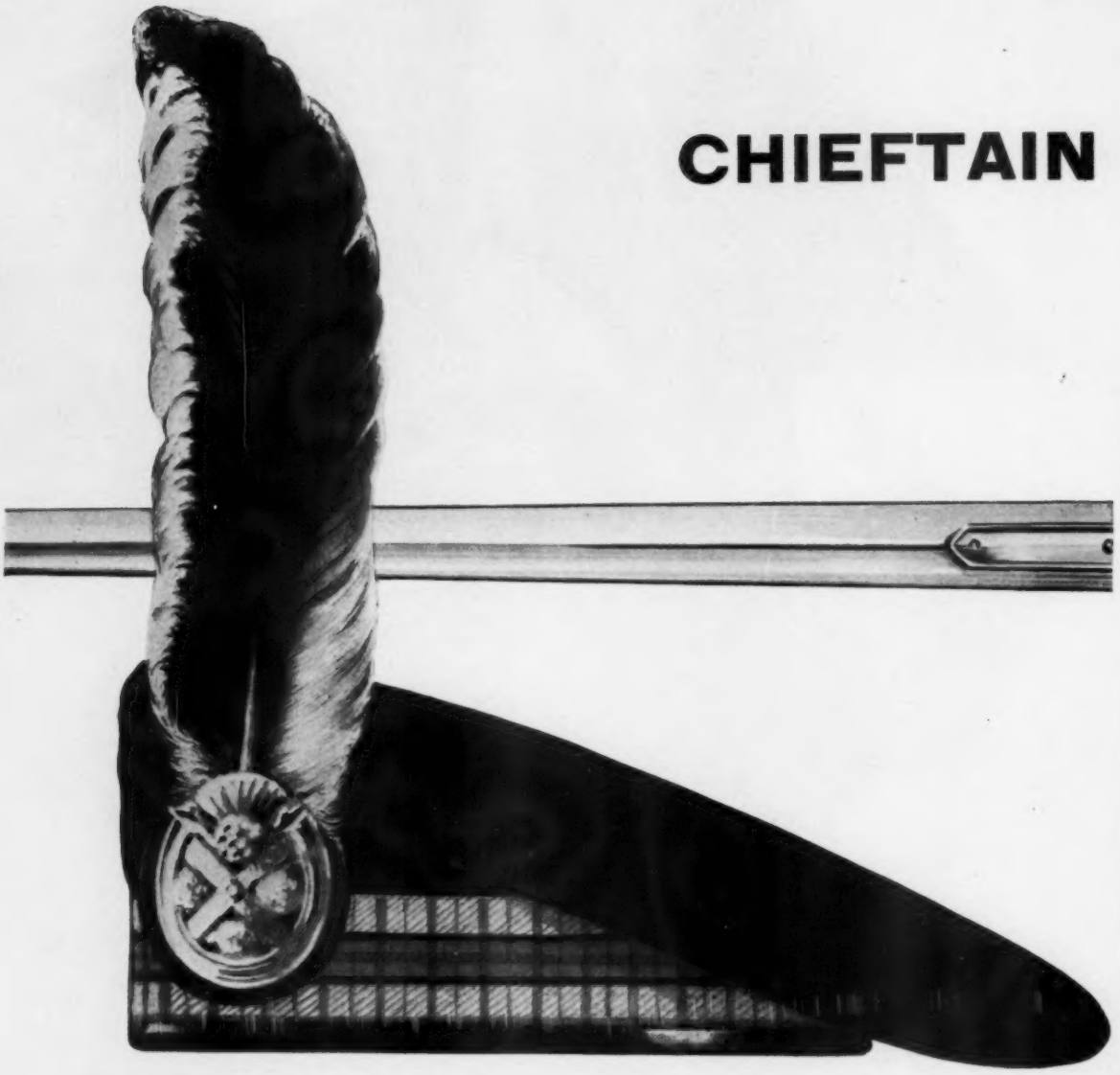

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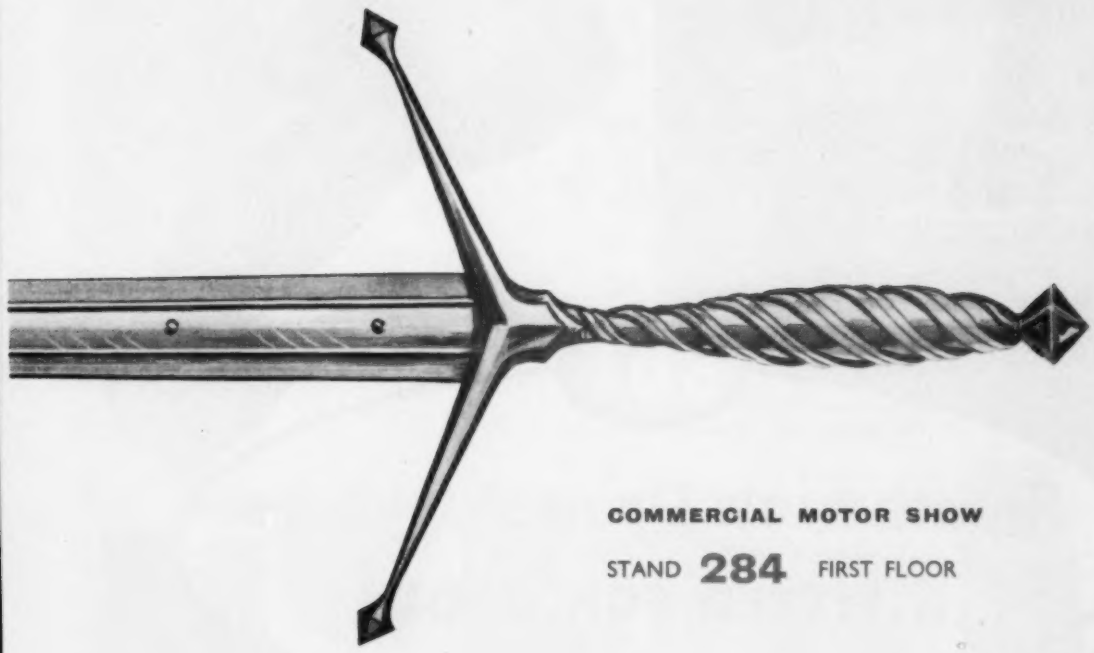

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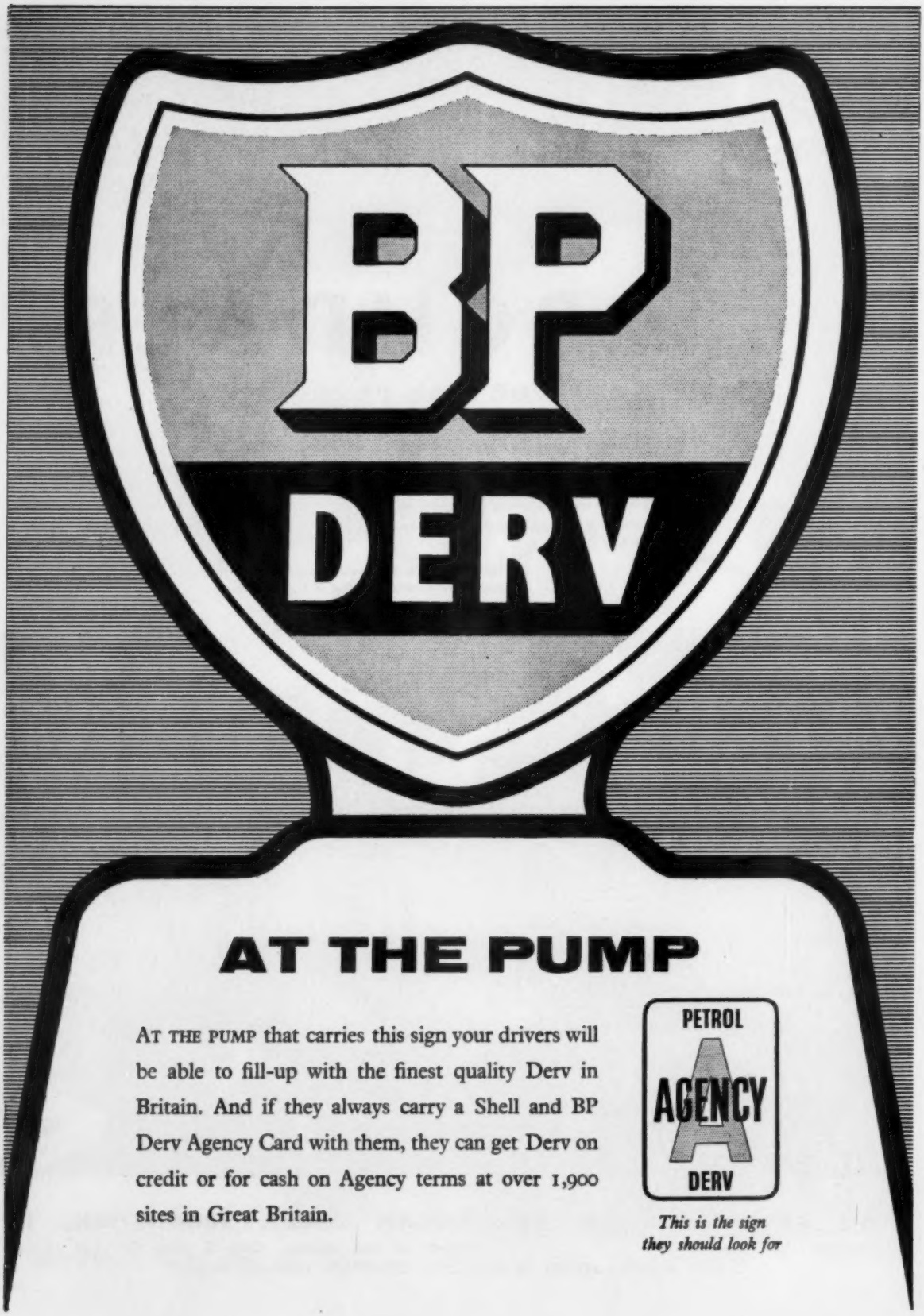

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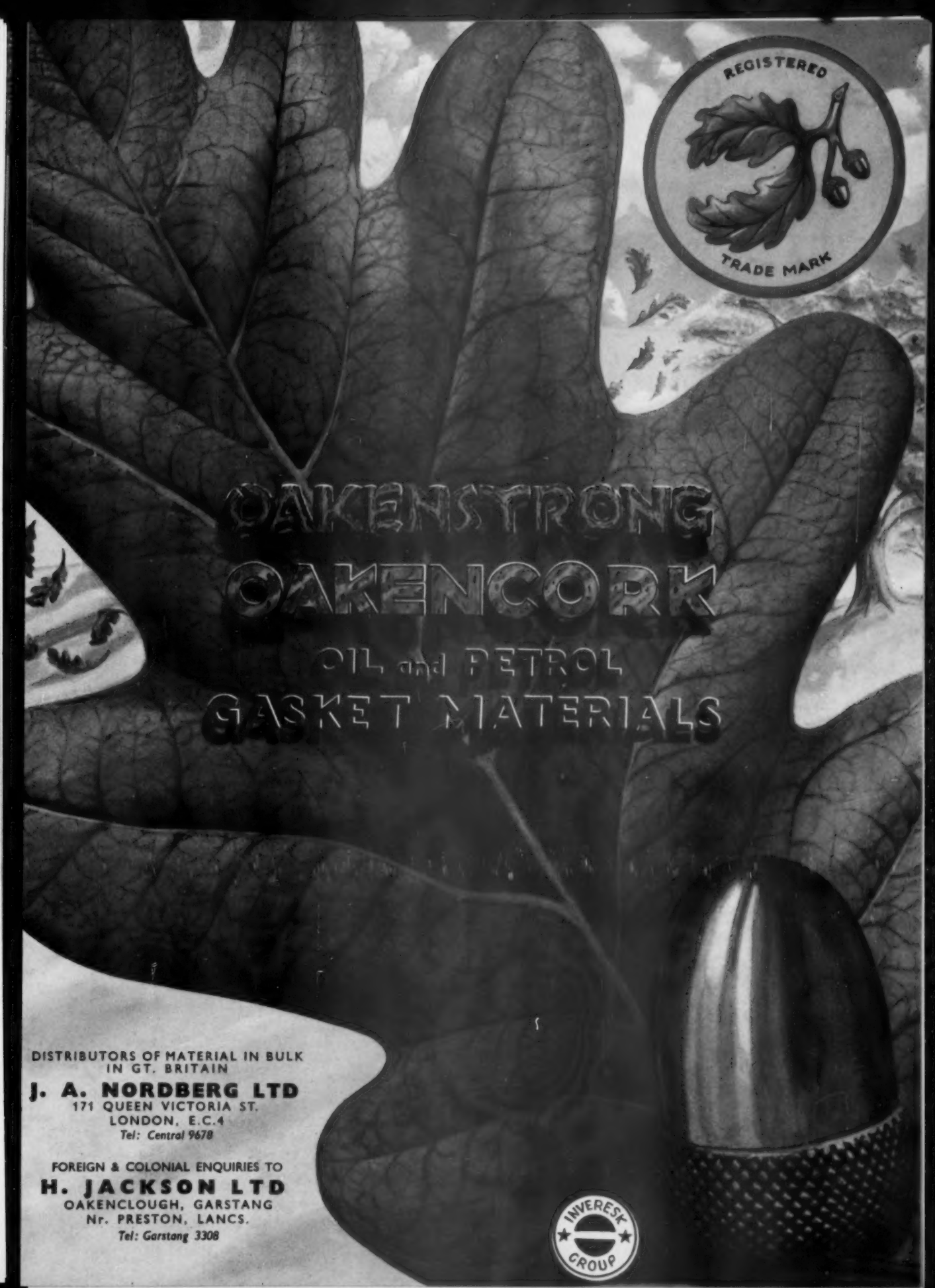
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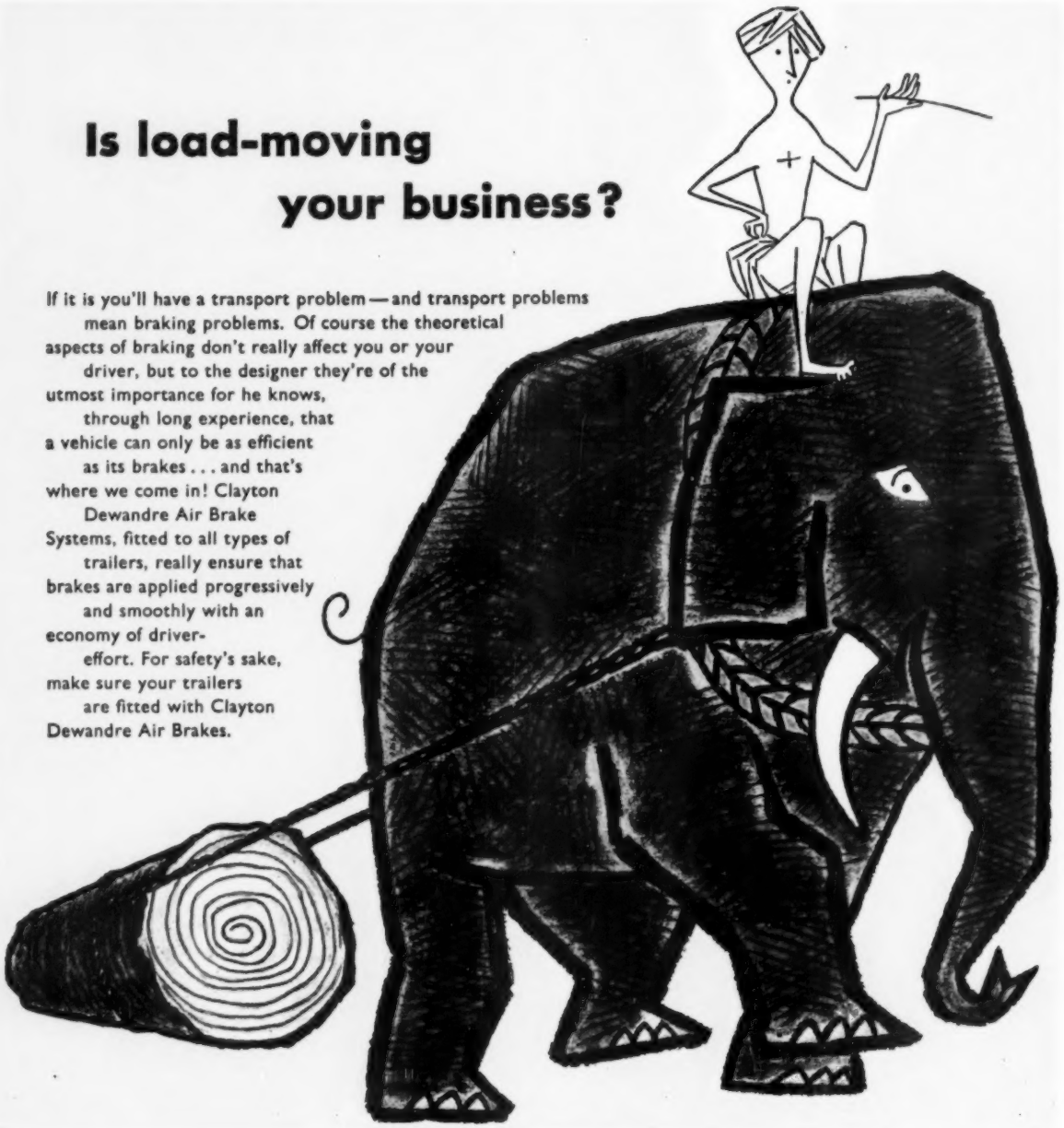

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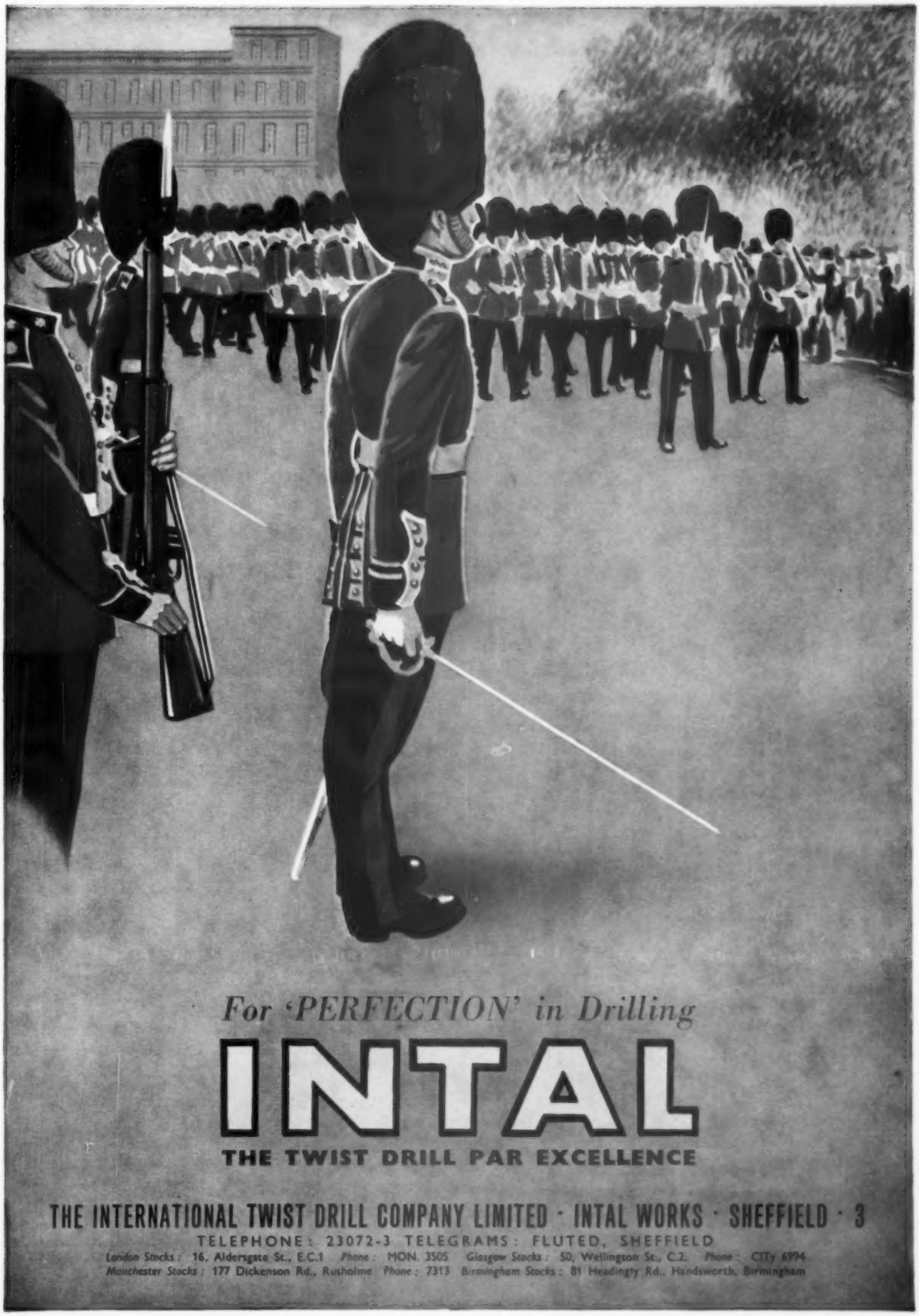
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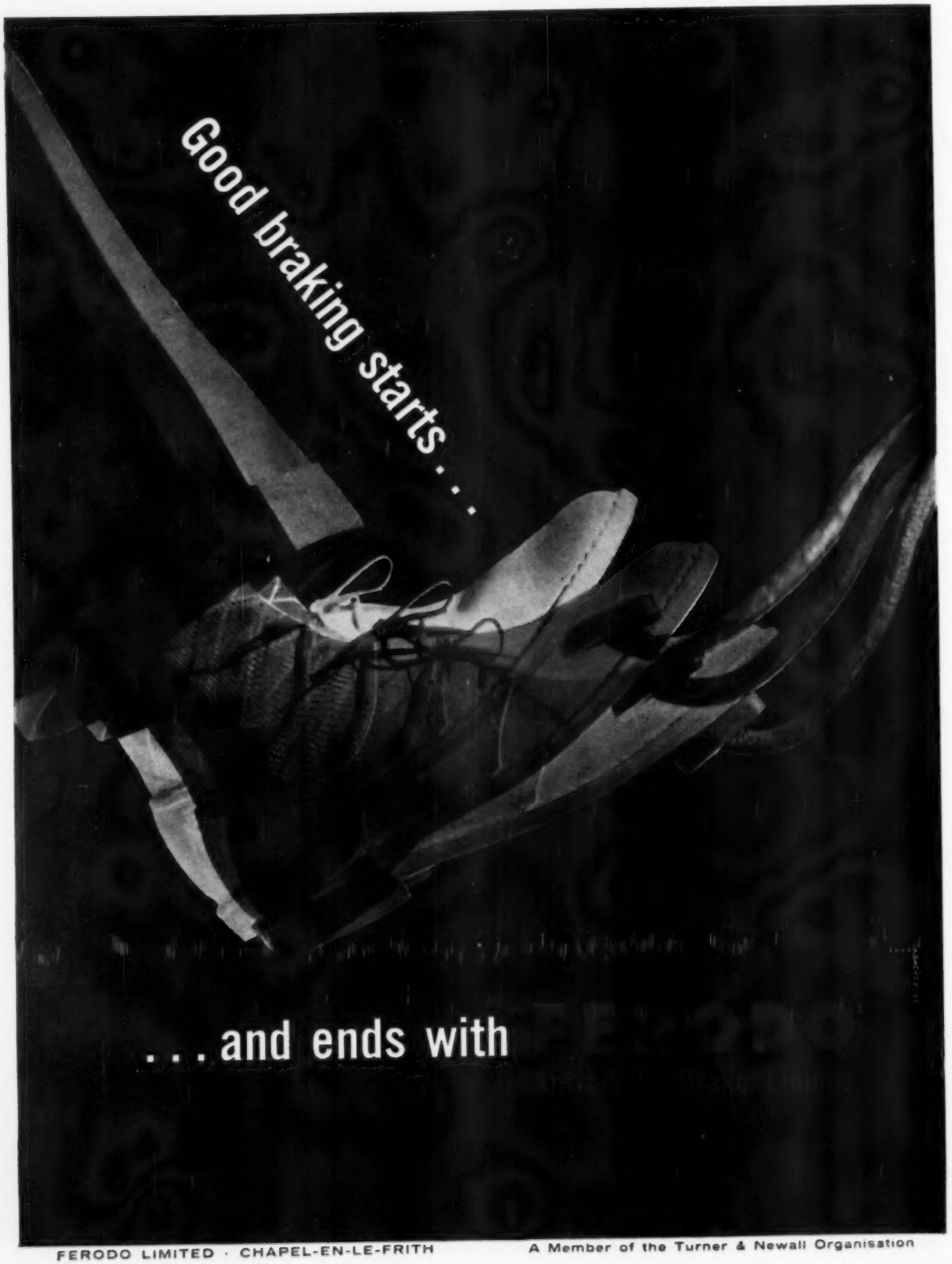

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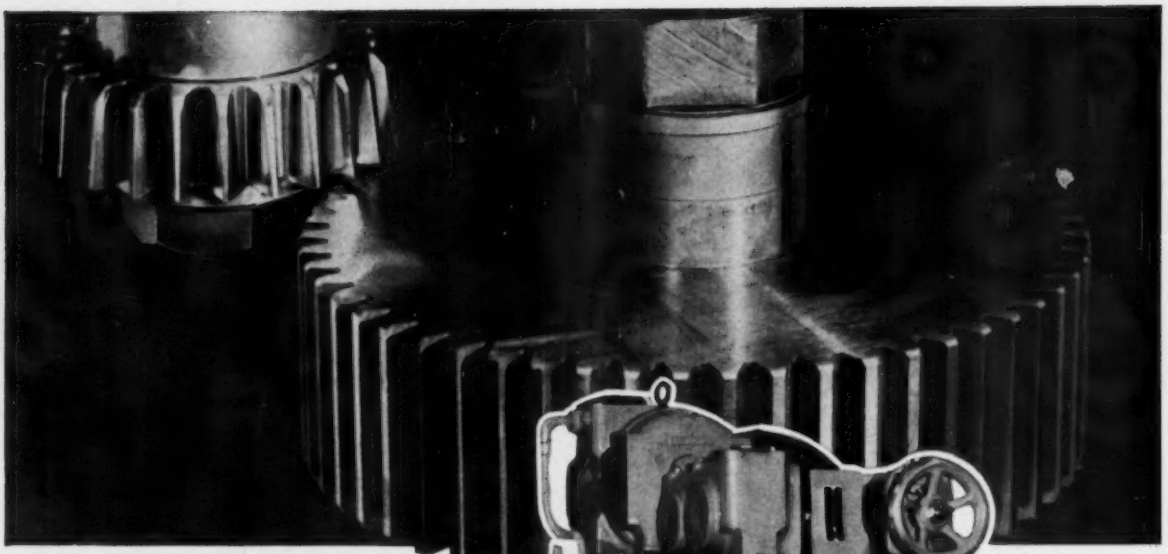

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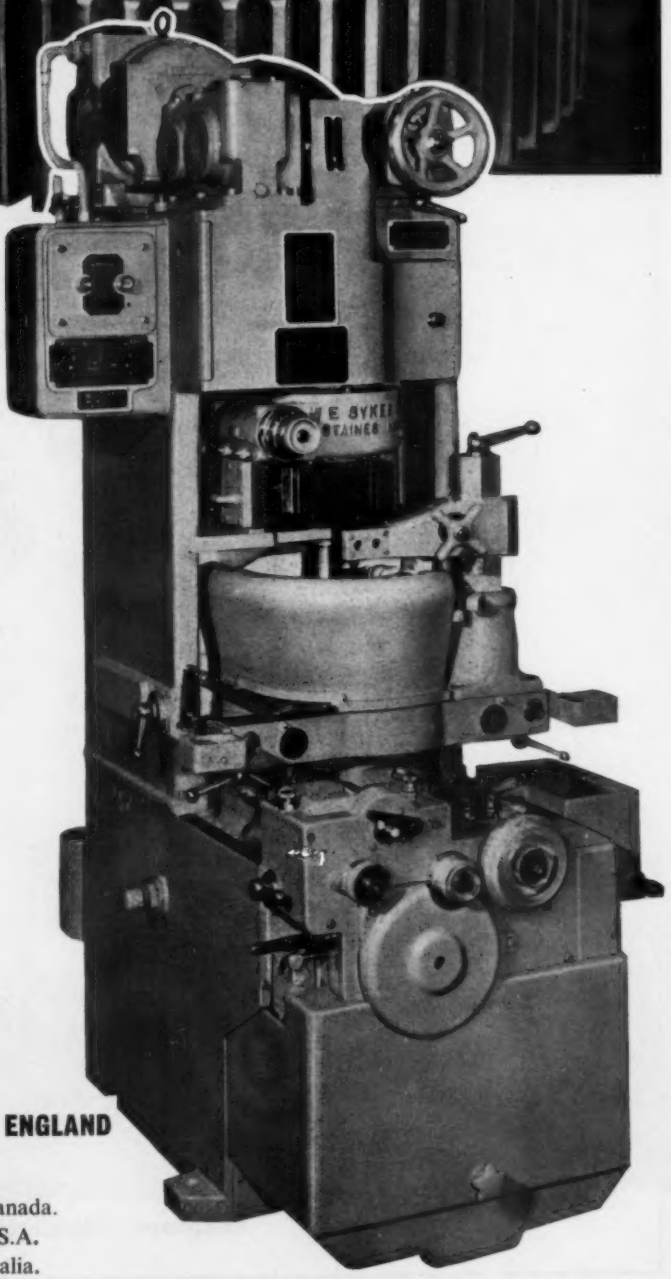

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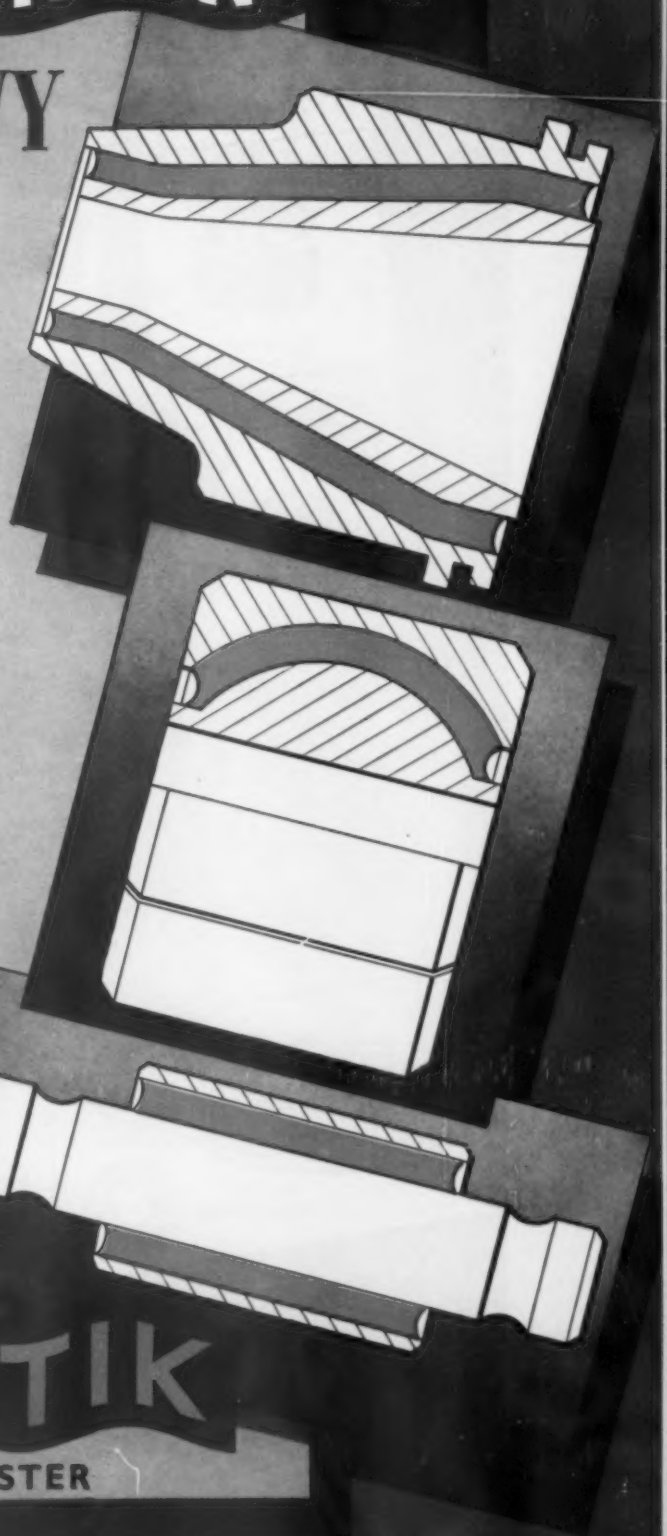

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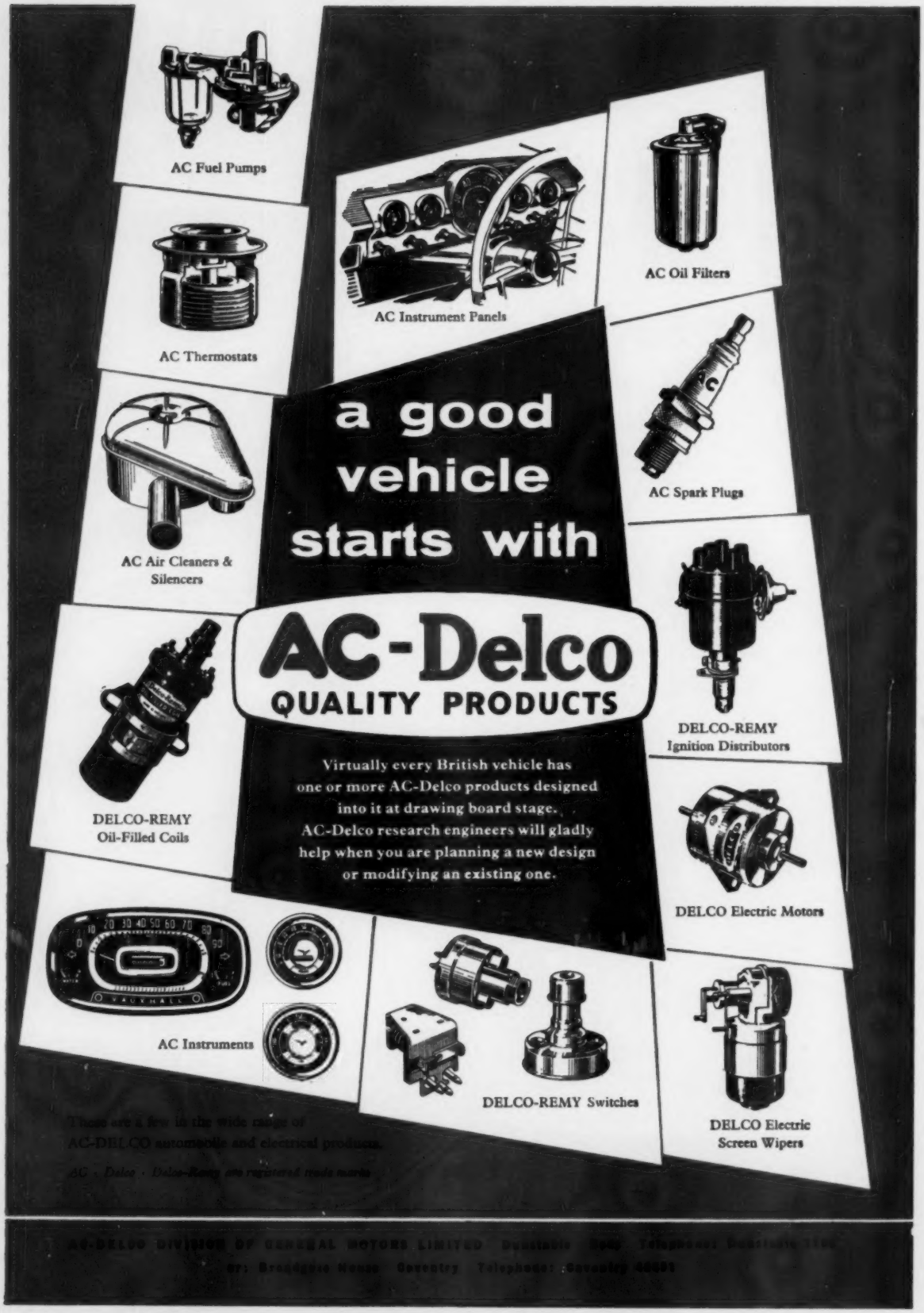
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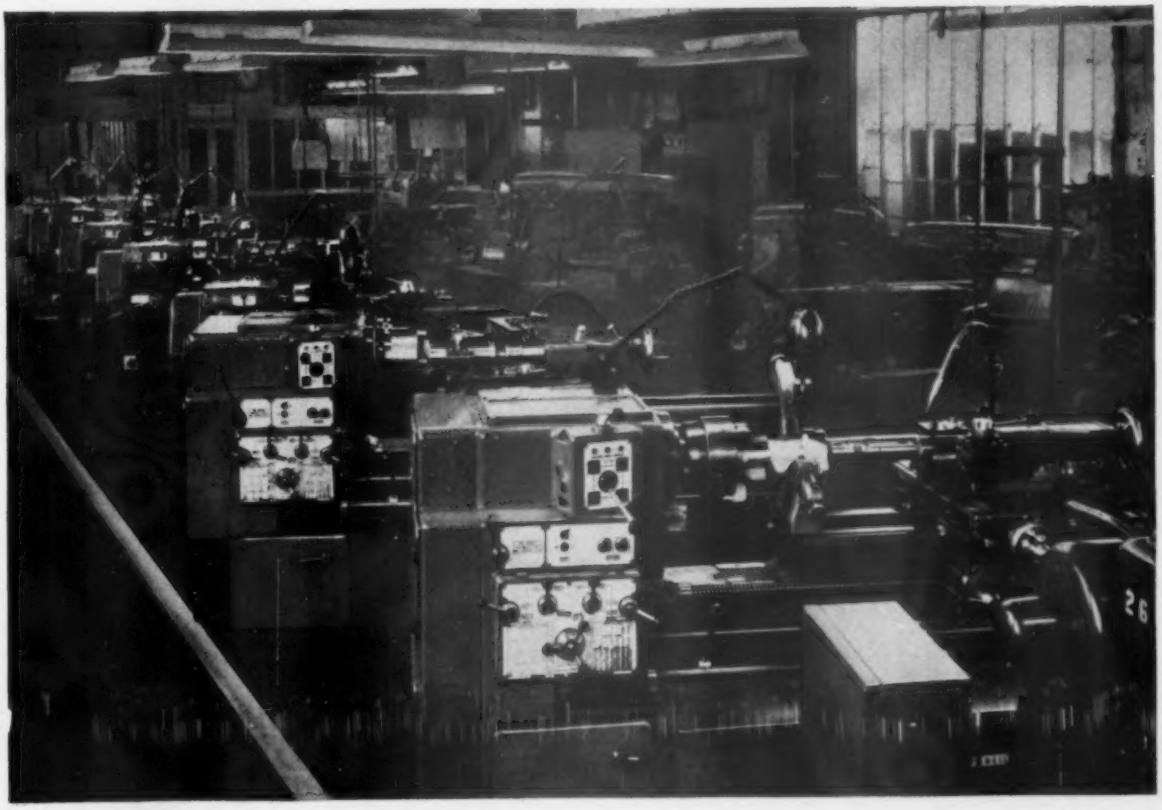

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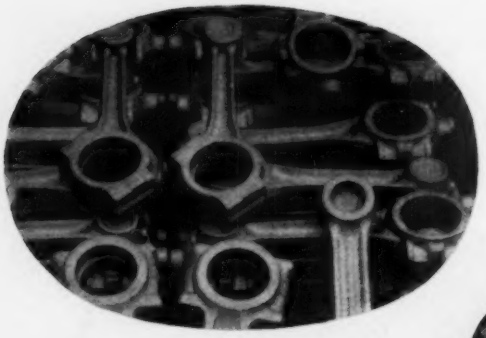

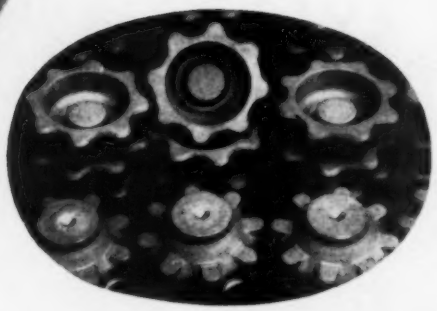

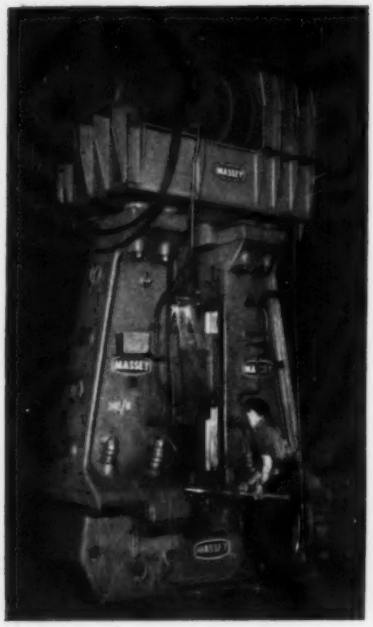

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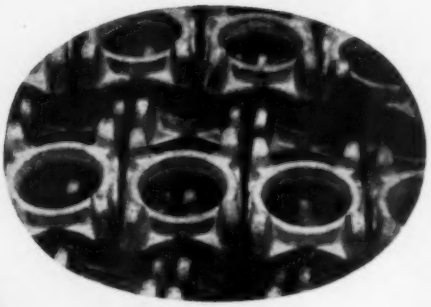

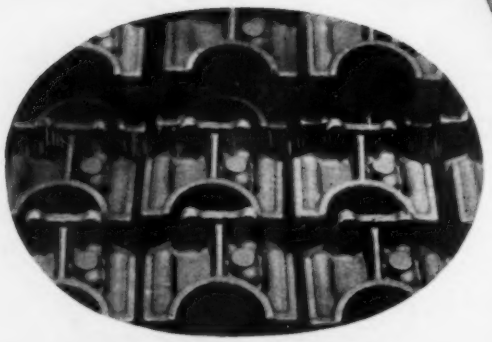

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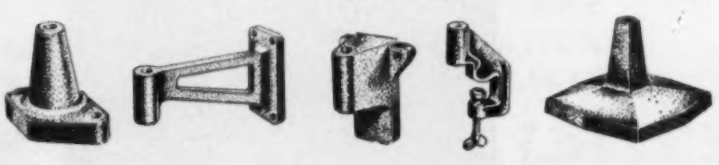

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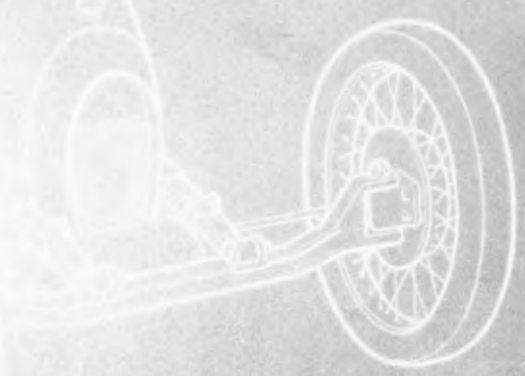
Silentbloc and oil-seal gaiters by Andre Rubber
especially interesting because Silentbloc and And
men have been associated with racing cars from t
very earliest days of motor sport. In 1909 T. B. And



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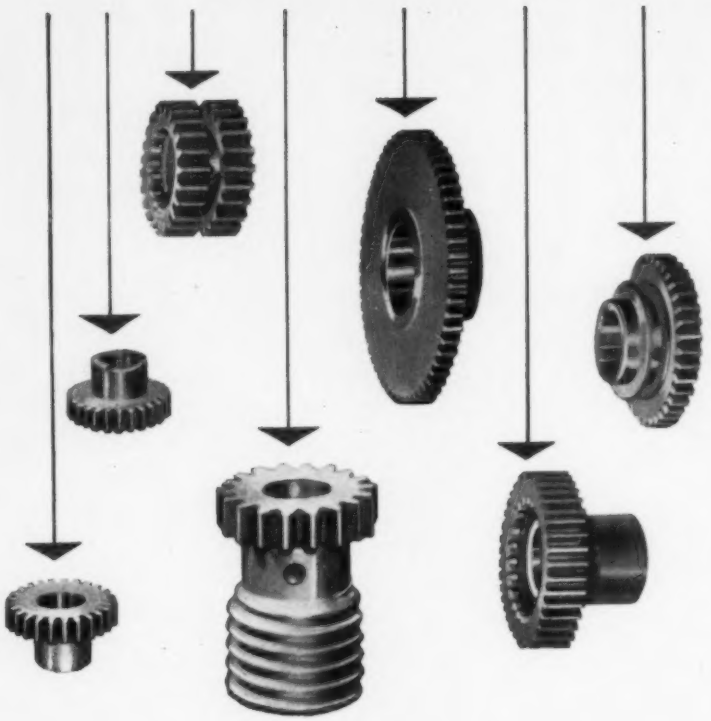

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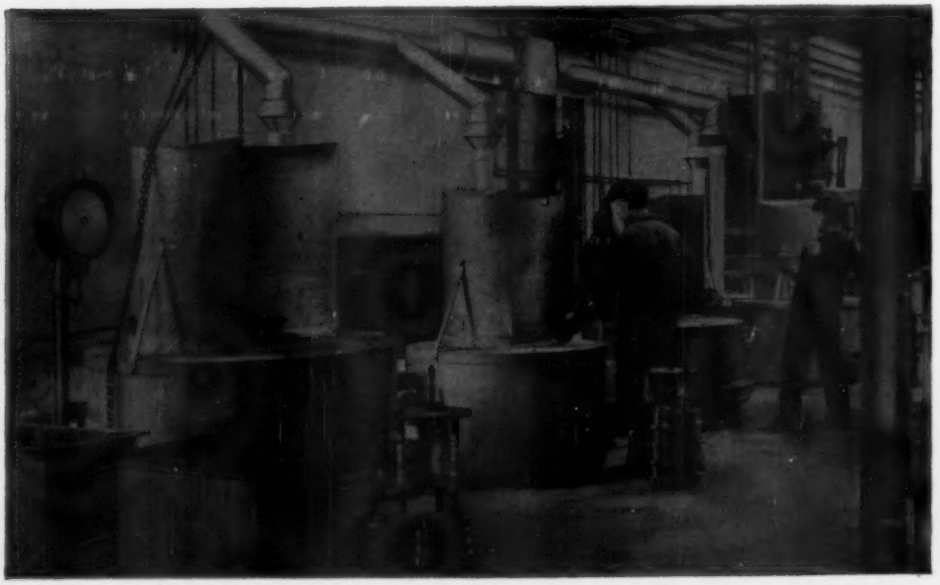
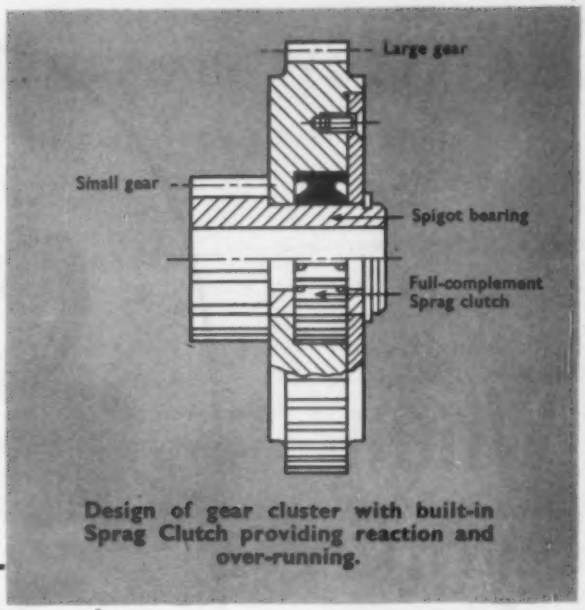


The Heat Treatment Shop—H. W. Ward & Co. Ltd., Worcester

ICI **IMPERIAL CHEMICAL INDUSTRIES LIMITED, LONDON, S.W.1**

CC191

for automotive applications



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A simple, compact unit for:

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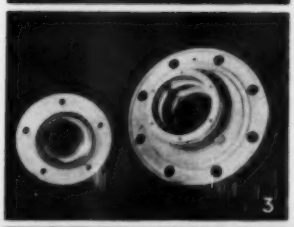
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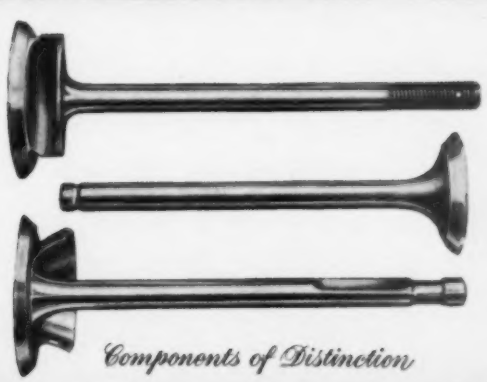

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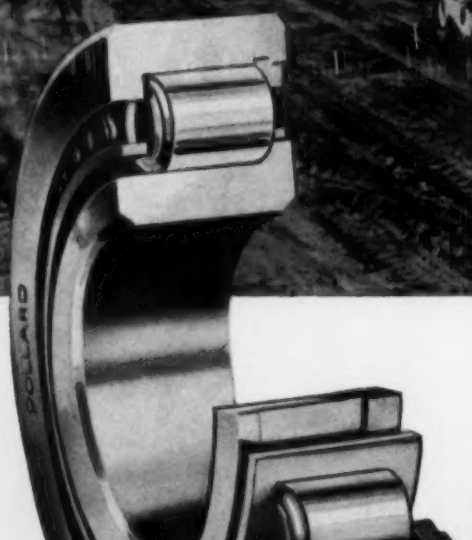

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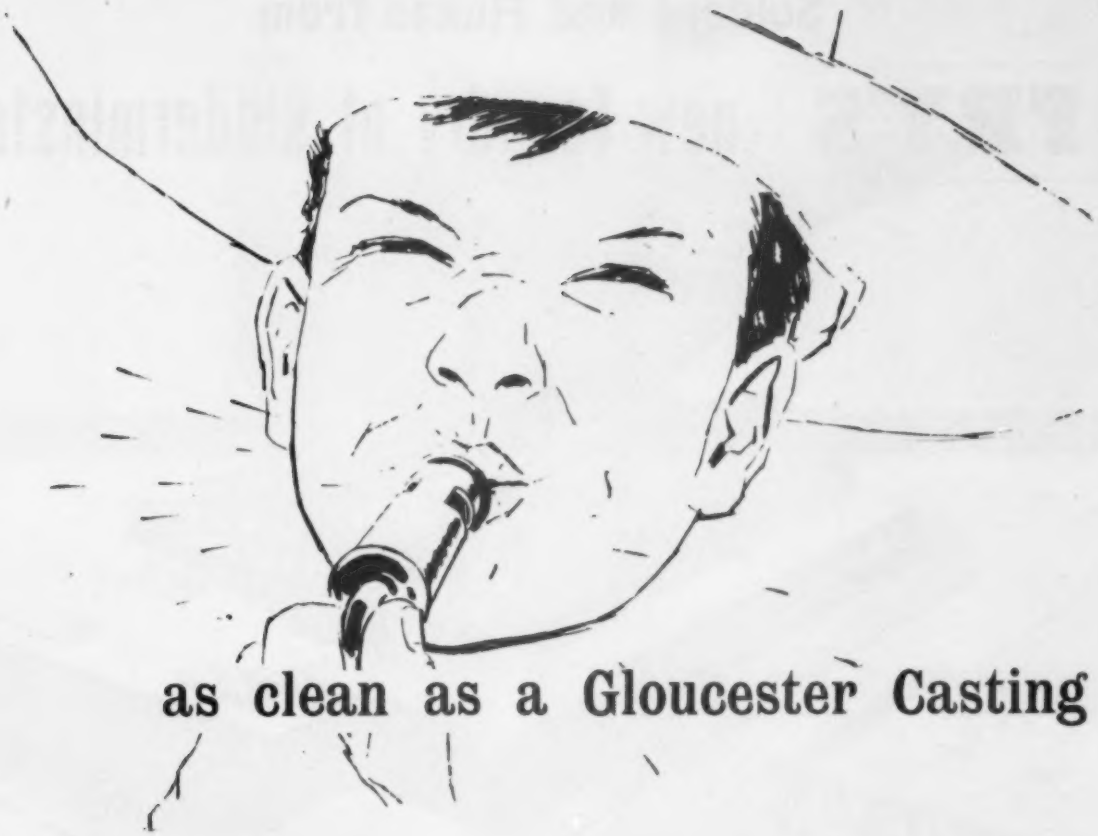

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Bearing bracket in Malleable Iron. Weight 3½lbs.

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Yield Point	12 tons	Yield Point	24 tons
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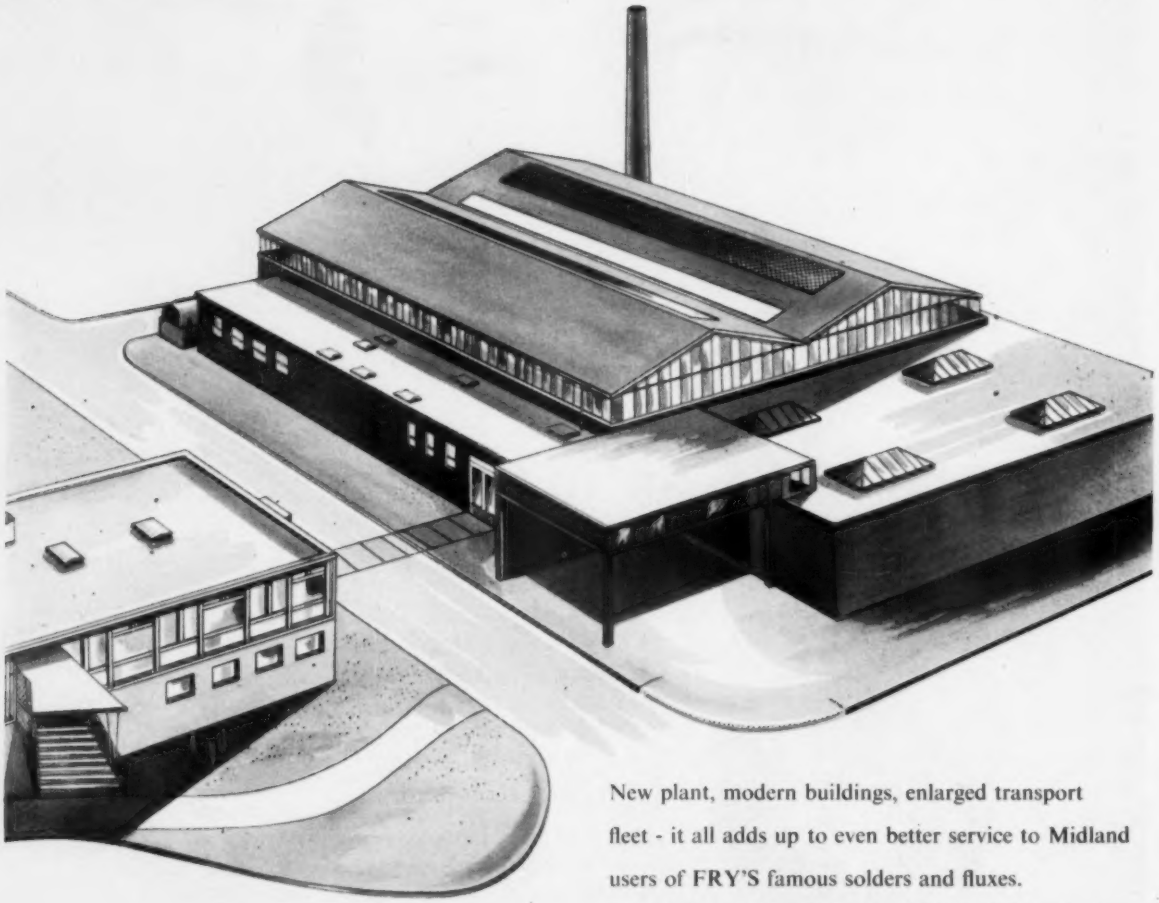

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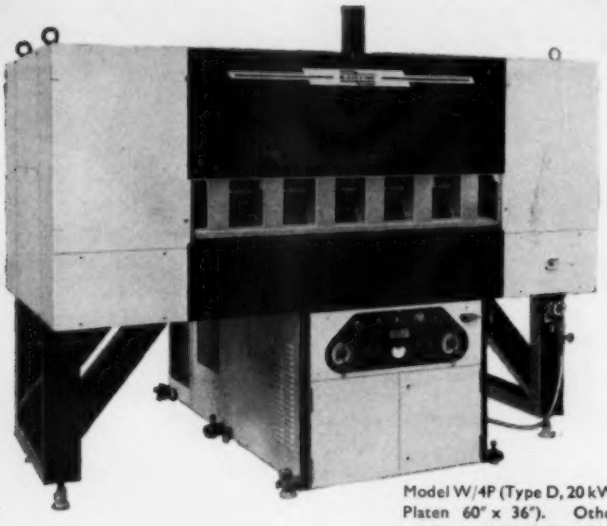

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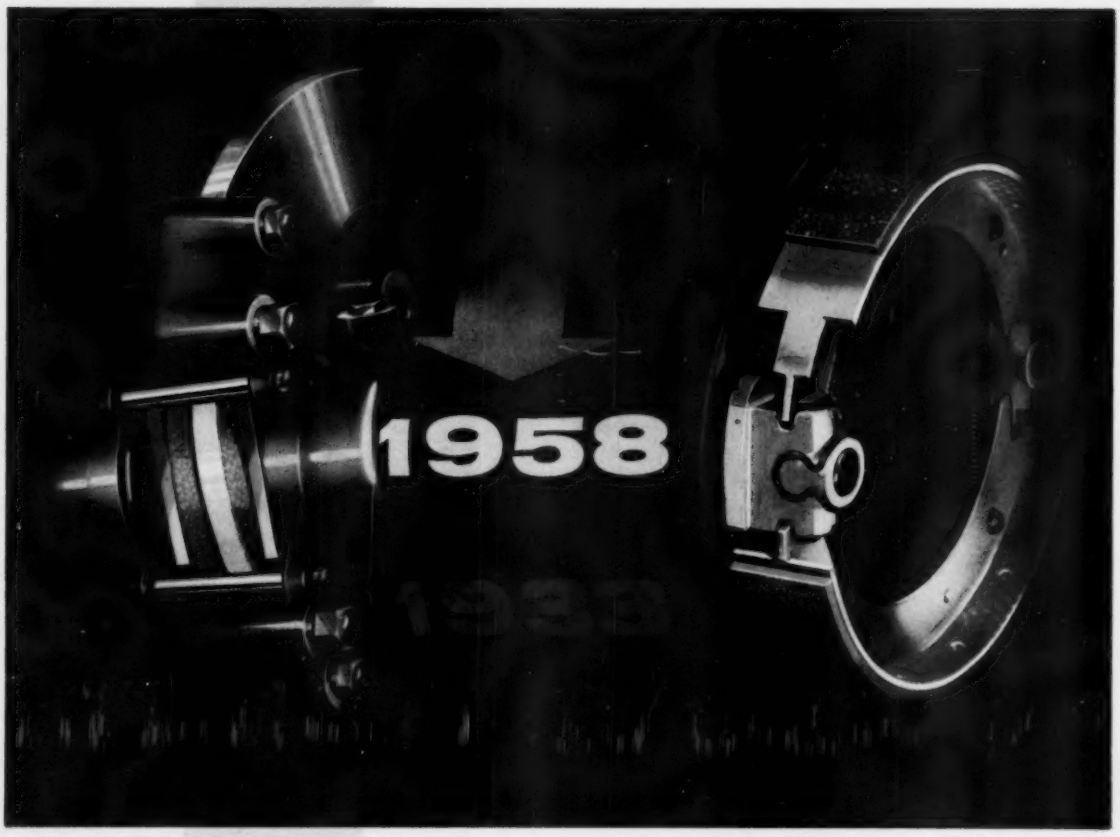

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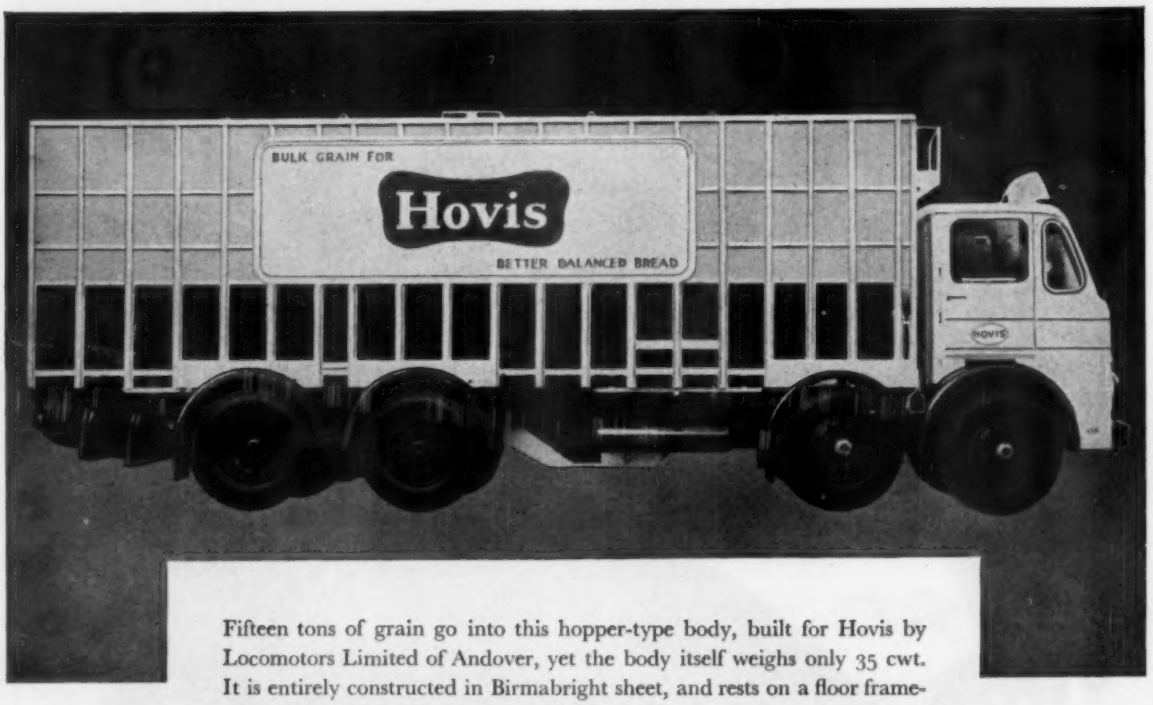
VB48

Hovis use BIRMABRIGHT for bulk grain transport



Fifteen tons of grain go into this hopper-type body, built for Hovis by Locomotors Limited of Andover, yet the body itself weighs only 35 cwt. It is entirely constructed in Birmabright sheet, and rests on a floor framework of Birmabright extruded sections.

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Do you know how Birmabright aluminium-magnesium alloys simplify bodywork construction, and how large a range of materials is available from stock? Why not ask us for full information?

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BH91

ASQUITH

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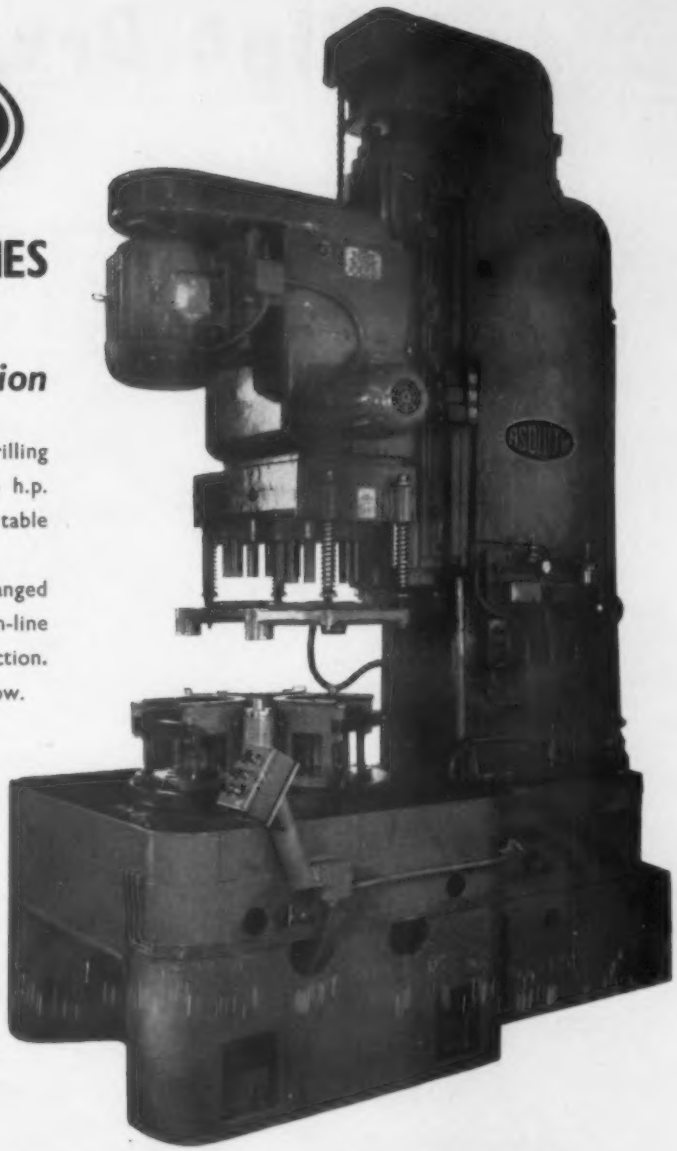
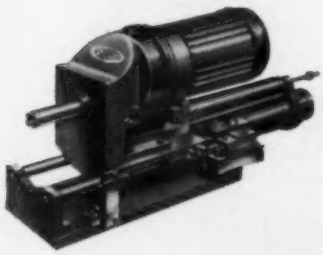
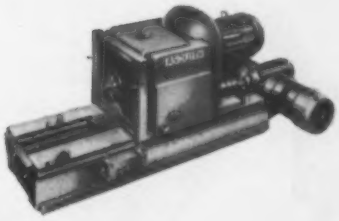


½h.p. AIR HYDRAULIC DRILLING UNIT



3/5h.p. SCREW FEED UNIT



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A302

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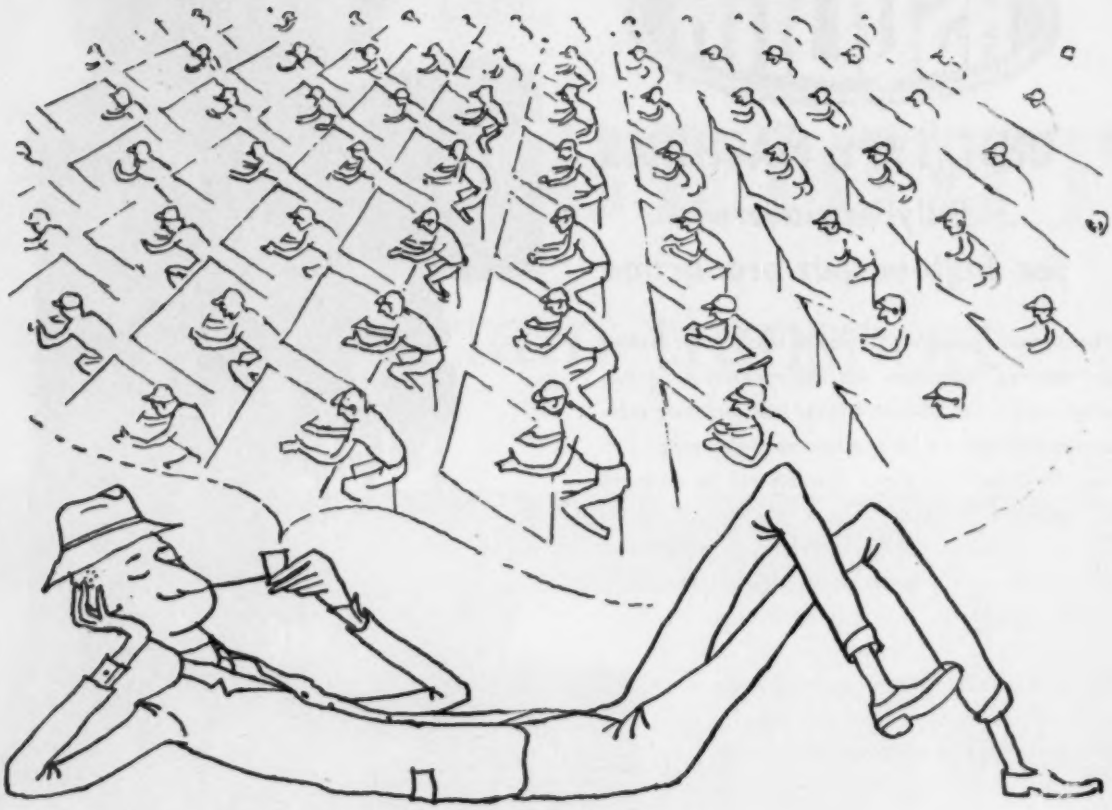

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Illustration shows ***'dag'*** *colloidal graphite being applied during engine-assembly at the works of the Standard Motor Co. Ltd., Coventry.*

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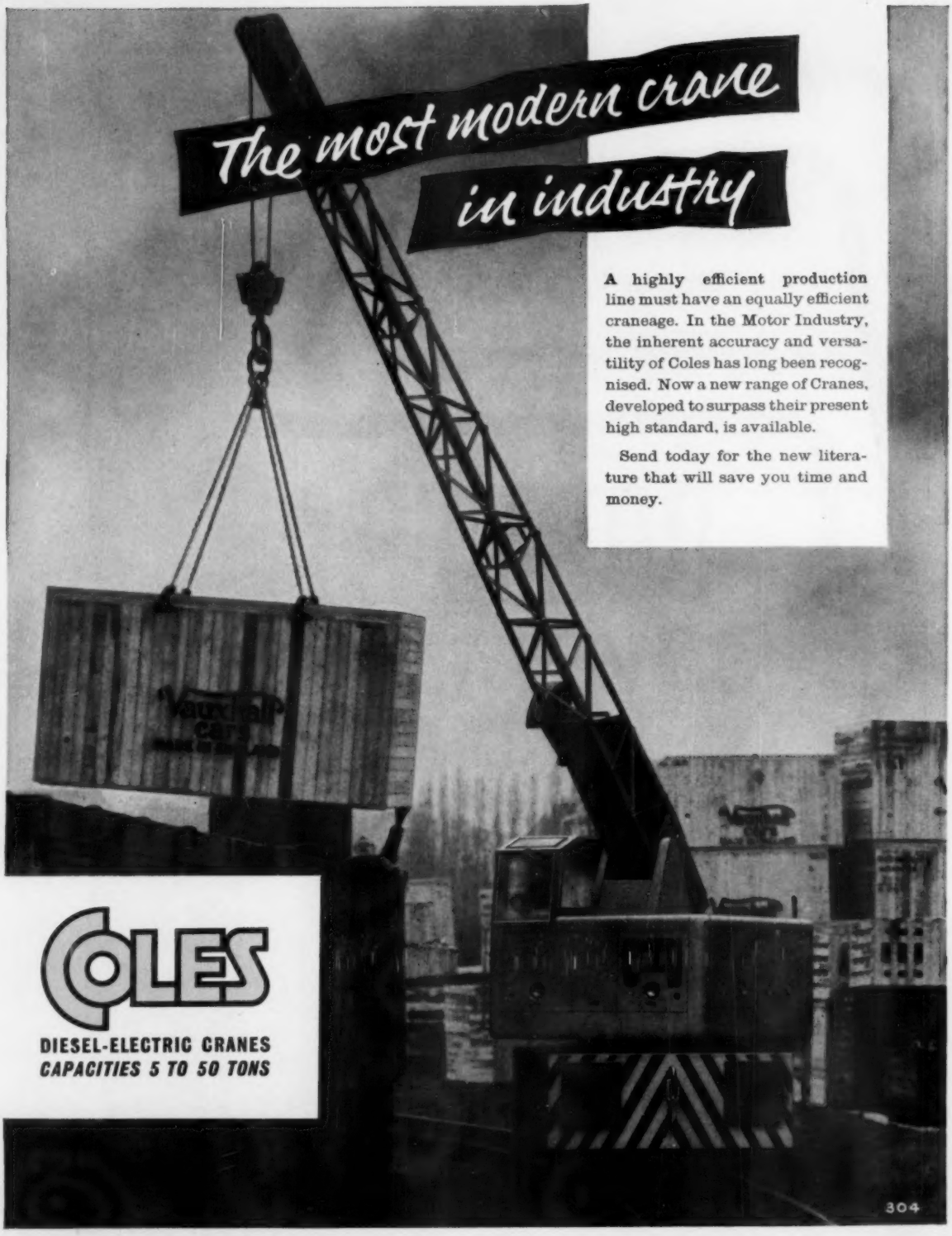
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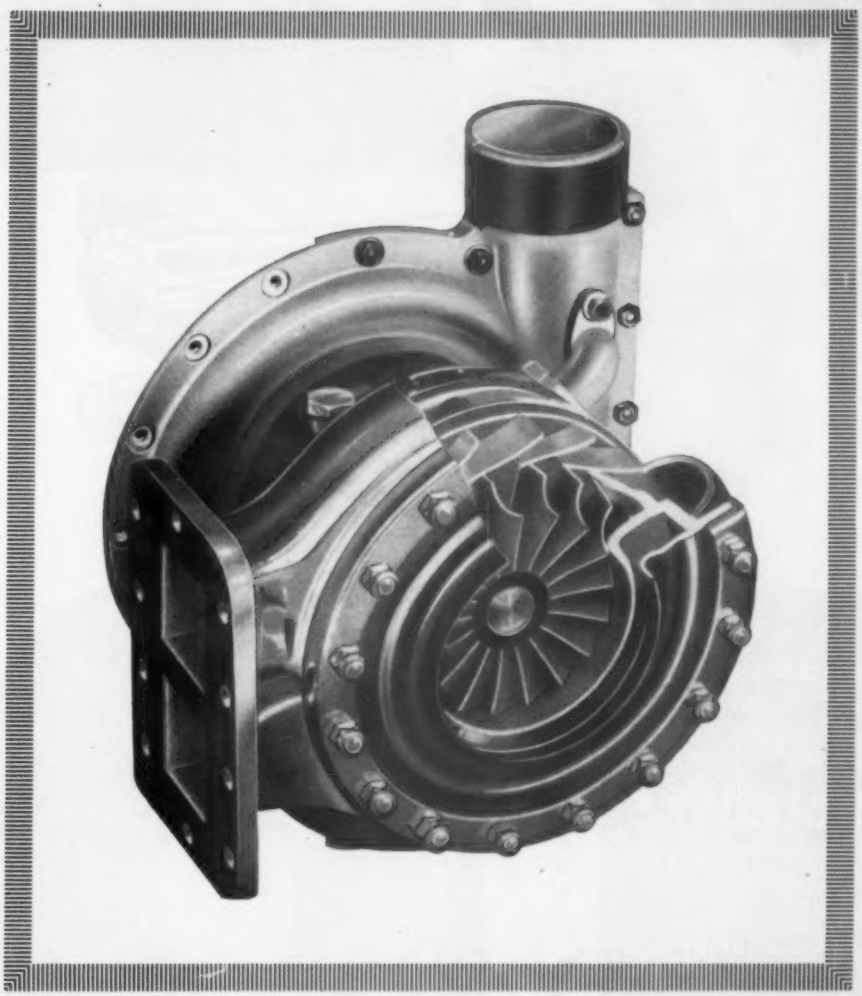

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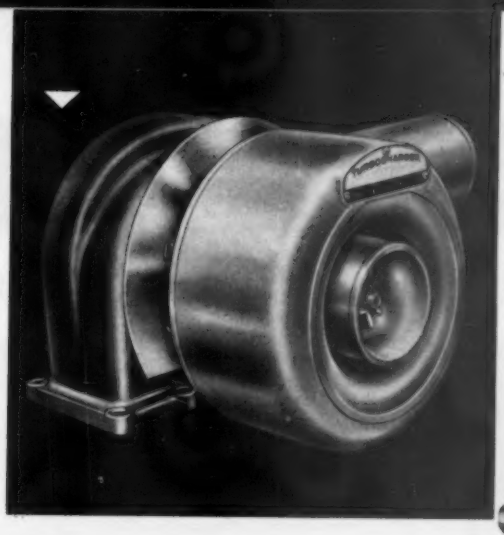

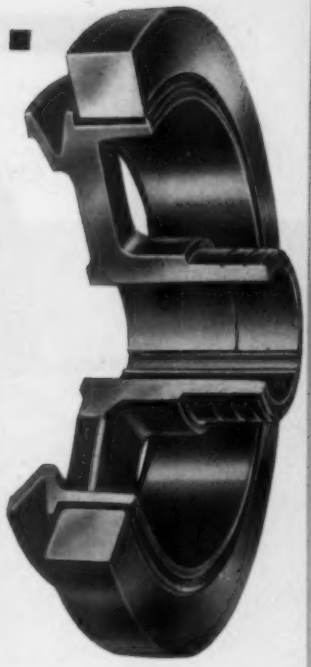

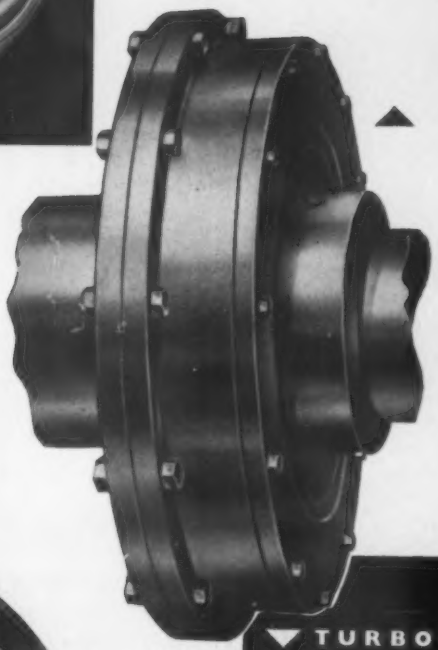

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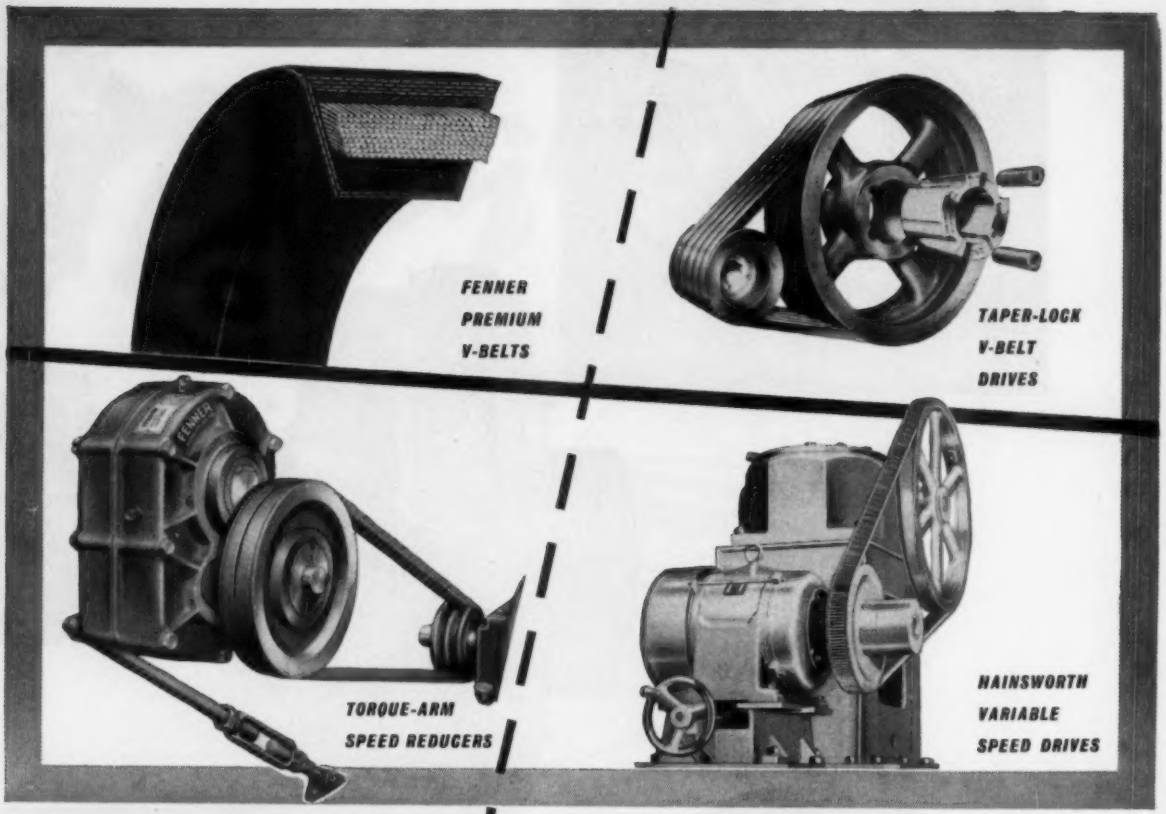

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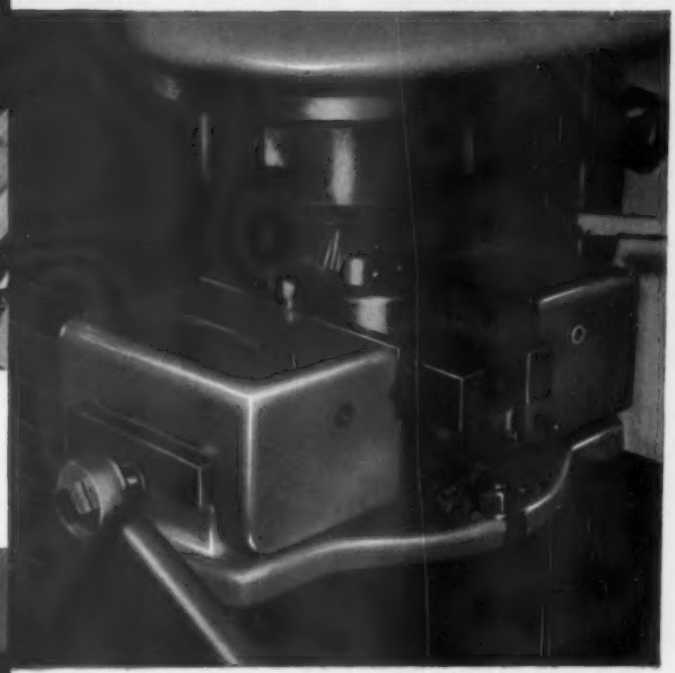

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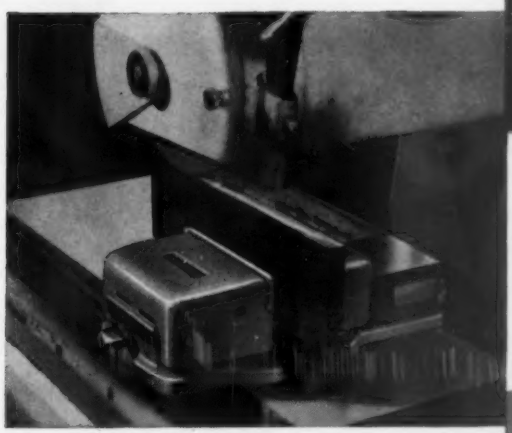

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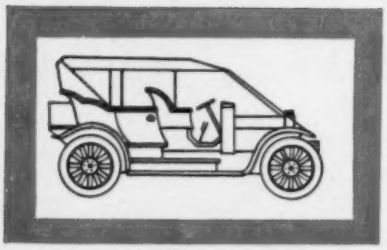

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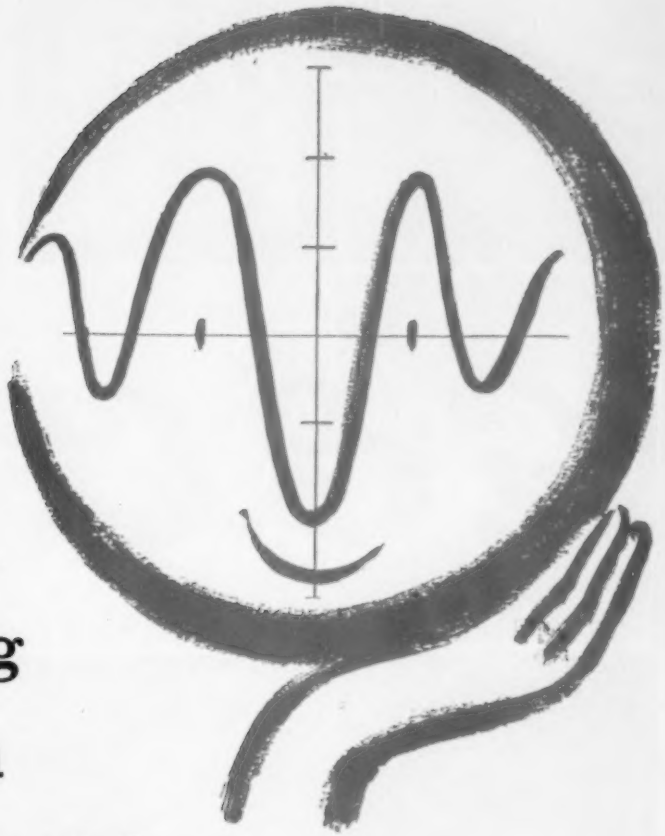

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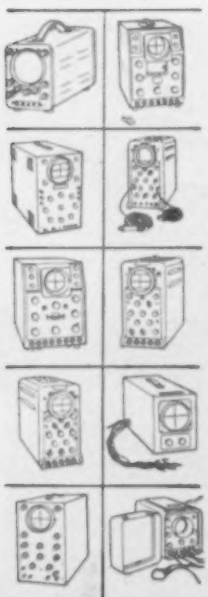

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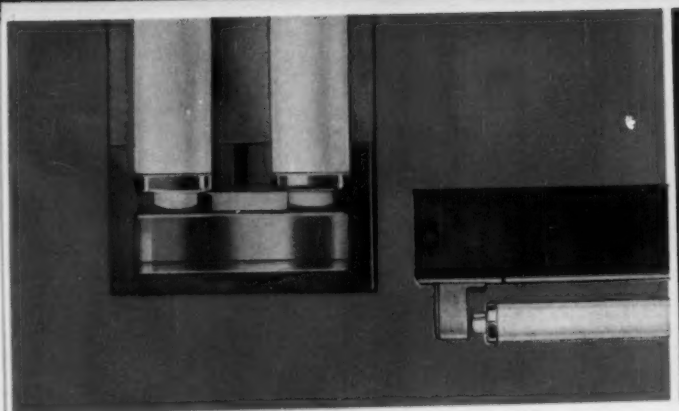

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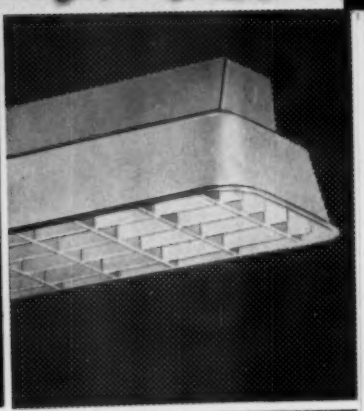

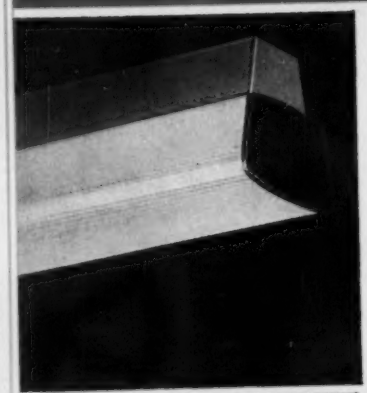

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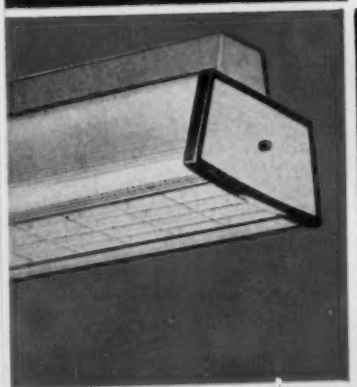

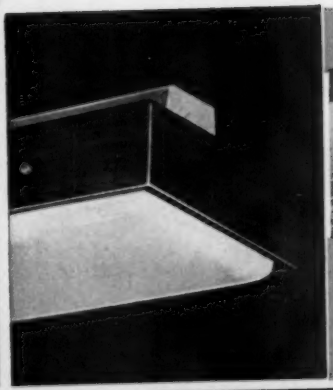

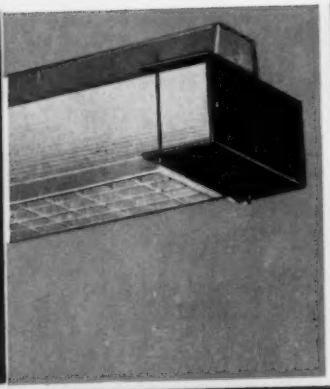

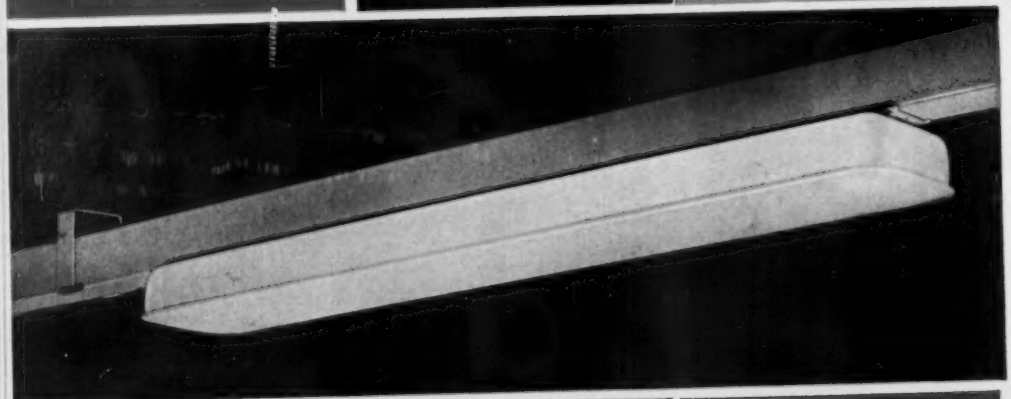

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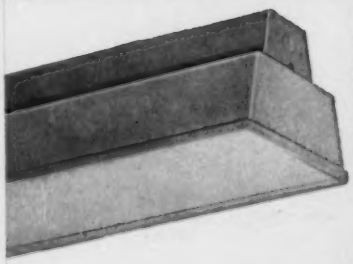

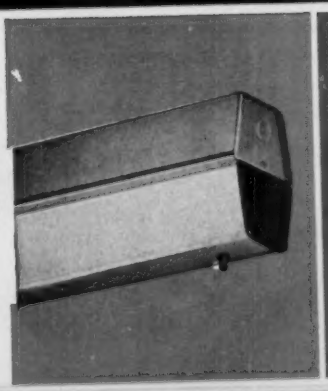

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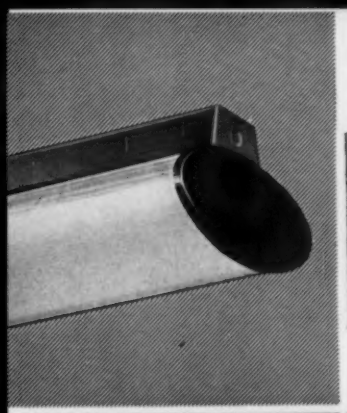

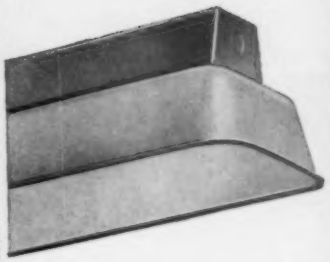

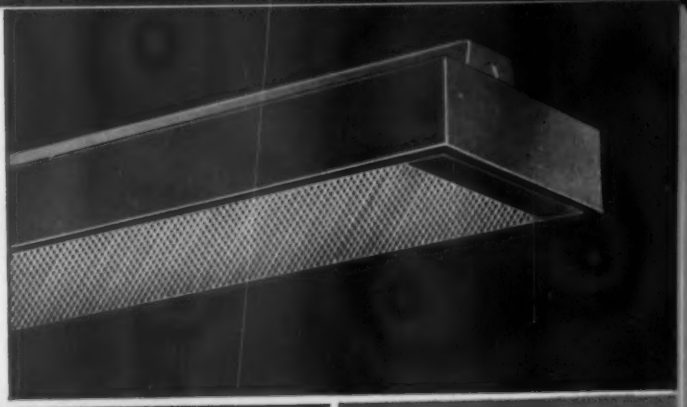

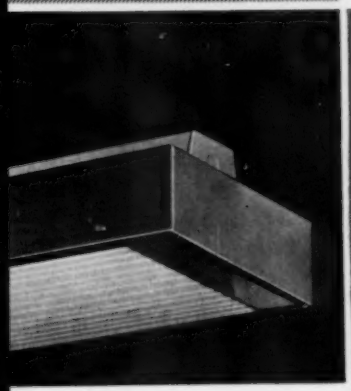

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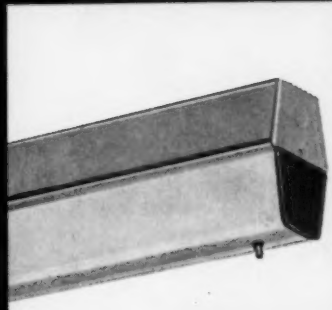

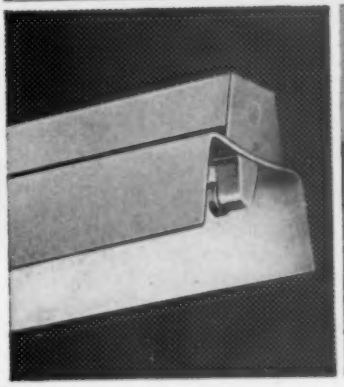

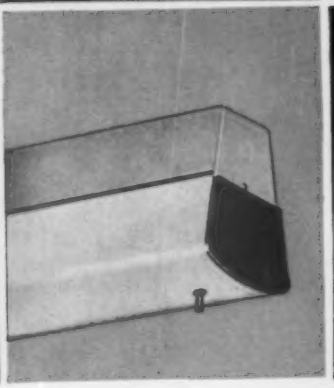

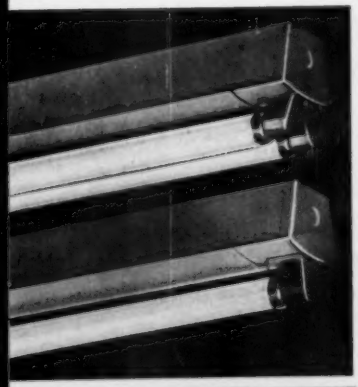

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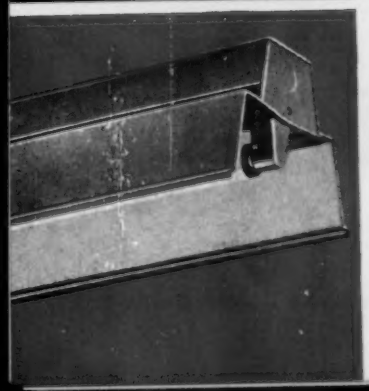

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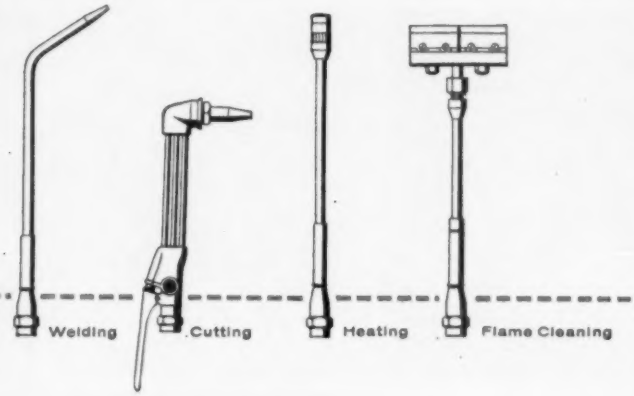

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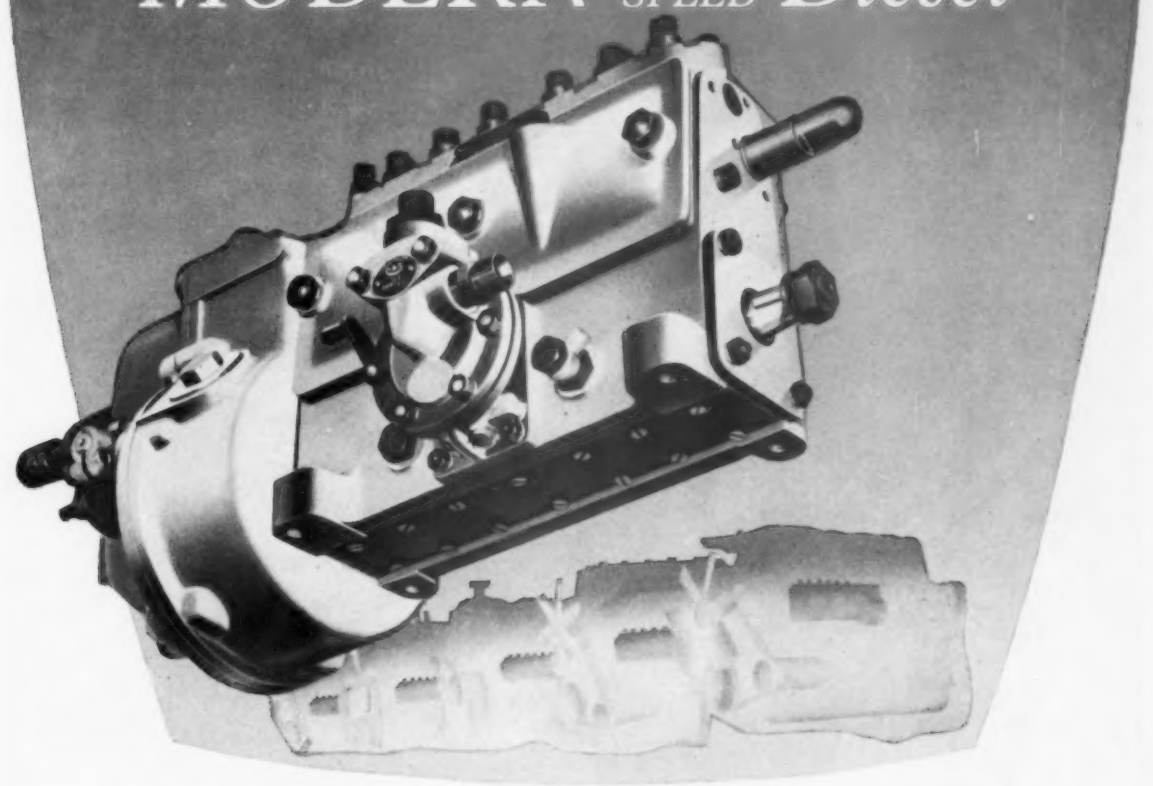

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The annual subscription inland and overseas is £3 0s 0d including the special number
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VOLUME 48 **NUMBER 10** **OCTOBER 1958**

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Diesel Engine Pistons

In recent years, considerable research and development has been devoted to the improvement of design and production techniques for pistons of compression ignition engines. As a result, it is now possible to operate engines at ratings which, ten years ago, would have been regarded as impossible without resort to oil-cooling of the pistons. Many engineers closely concerned with the design and development of pistons for this type of engine consider that, so far as the mechanical aspect of design is concerned, the ultimate limit of progress has now almost been reached. Provided that accurate data on the engine performance and operating conditions are available, it is possible to manufacture, directly from the initial drawings, a piston that will function satisfactorily without further alteration. In fact, piston design has almost developed into an exact science.

However, in view of the continued trend in engine design towards raising power: weight ratios, by reducing the centre-to-centre lengths of the connecting rods, and consequently employing pistons with smaller length: diameter ratios, it is obvious that still further improvements are needed in piston design. In what direction, then, can further progress be made? It would appear that the solution to the problem of how to obtain even better performance from pistons in the future lies in the metallurgical field.

For years, it was almost universal practice to use Y-alloy for pistons intended for application to highly rated diesel engines. Unfortunately, this material has a high coefficient of expansion. For example, the figure generally quoted for Y-alloy is in the order of 0·000025 per deg C, whereas some of the latest materials, such as 22-24 per cent silicon, hyper-eutectic alloy have a coefficient of expansion of about 0·000016 per deg C. Pistons made of the latter material can, of course, be designed with cold clearances comparable with those of similar components made of cast iron, the coefficient of expansion of which is approximately 0·000011 per deg C. The development of these new alloys is of considerable significance because, with the relatively short pistons that are now coming into use, there is a danger of piston rocking, particularly during idling, at low temperature conditions of operation. This trouble, which has been most noticeable with the shorter pistons made of Y-alloy, has an adverse effect, of course, so far as ring groove wear is concerned. Employment of the new material, with a high silicon content, generally obviates this difficulty, even where the length: diameter ratio of the piston is less than 1:1.

Although for many years the use of these silicon alloys has been opposed, because of their relatively low tensile strength, this argument is losing force as a result of recent developments. By the employment of modern methods of casting and heat treatment, it is possible appreciably to improve the properties of these materials, especially at the normal temperatures of operation in the engine. Metallurgists have also studied carefully the effects of slight variations in the chemical composition of these alloys. As a result of this work, hyper-eutectic alloys have been produced with tensile strengths, at temperatures above 300 deg C, that are higher than those of the majority of the currently widely accepted grades of piston alloys. Further progress is now being made by anodically treating the ring grooves to obtain locally a surface hardness in the region of 800-900 Brinell. This is expected to increase the life of the piston by 10-15 per cent, and will bring the life expectancy of this component between overhauls into line with that of other parts of the engine.

Some years ago, especially before the 1939-45 War, the anodic treatment of pistons was viewed with disfavour. This was because, despite the fact that considerable research and development work had been carried out with a view to obtaining better results with pistons treated in this manner, it was found to be difficult to obviate the tendency of the coating to flake during service. Failures of this type were generally attributable to the high rates of expansion of the alloys then used, but it has now been established that, with the alloys of high silicon content, the anodic film does not easily flake. In fact, it would appear that, within practical limits, the higher the silicon content, the better is the adhesion of the film. With alloys containing 24-25 per cent silicon, this surface treatment gives a finish that is velvet-like in appearance, but nevertheless is extremely hard. It also has good oil-retaining properties.

From the point of view of the foundryman, it is unfortunate that as the silicon content is increased, so are the casting problems. One of the greatest difficulties is to keep the silicon in uniform suspension. To do this, it may be necessary to accelerate cooling, since segregation occurs even during the solidification stage. Similar problems have been solved by careful investigation and development, and there can be little doubt that the difficulties associated with the production of these alloys will be overcome.

The Austin Gipsy

Part II: Engine, Clutch, Gearbox, Transfer Box and Final Drive Units

EITHER a petrol or a diesel engine can be fitted, to suit the customers' requirements. Apart from the fuel economy feature of the diesel unit, each of these two engines is comparable so far as performance is concerned. The petrol engine is slightly more powerful, and yields a correspondingly better road performance. Originally it was developed before the Second World War, but was only put into production, for the Austin Sixteen car, immediately after the War. Since that time, it has gone from strength to strength, proving outstandingly versatile and adaptable. With remarkably little modification it has broken speed and endurance records, and has been used to power industrial equipment of all kinds.

With a compression ratio of 6·8 : 1 the petrol engine is rated at 62 b.h.p. at 4,100 r.p.m. Although the peak torque is quoted as 110 lb-ft at a speed of 1,500 r.p.m., this figure falls barely perceptibly as the speed rises to 2,200 r.p.m. As is to be expected, on the diesel engine, the flat part of the torque curve extends even further over the speed range, but it rises only a matter of 3 lb-ft from 1,500 r.p.m. to the peak figure of 89 lb-ft at 2,800, falling thereafter to 83 lb-ft at the maximum speed of 3,500 r.p.m., at which the power peak of 55 b.h.p. is reached. The relatively high-speed operation of the diesel unit is made possible by the Ricardo Comet III type of combustion chamber. Since its introduction in 1954, this diesel engine has gained an excellent reputation, being used as the standard power unit for many of the London taxis.

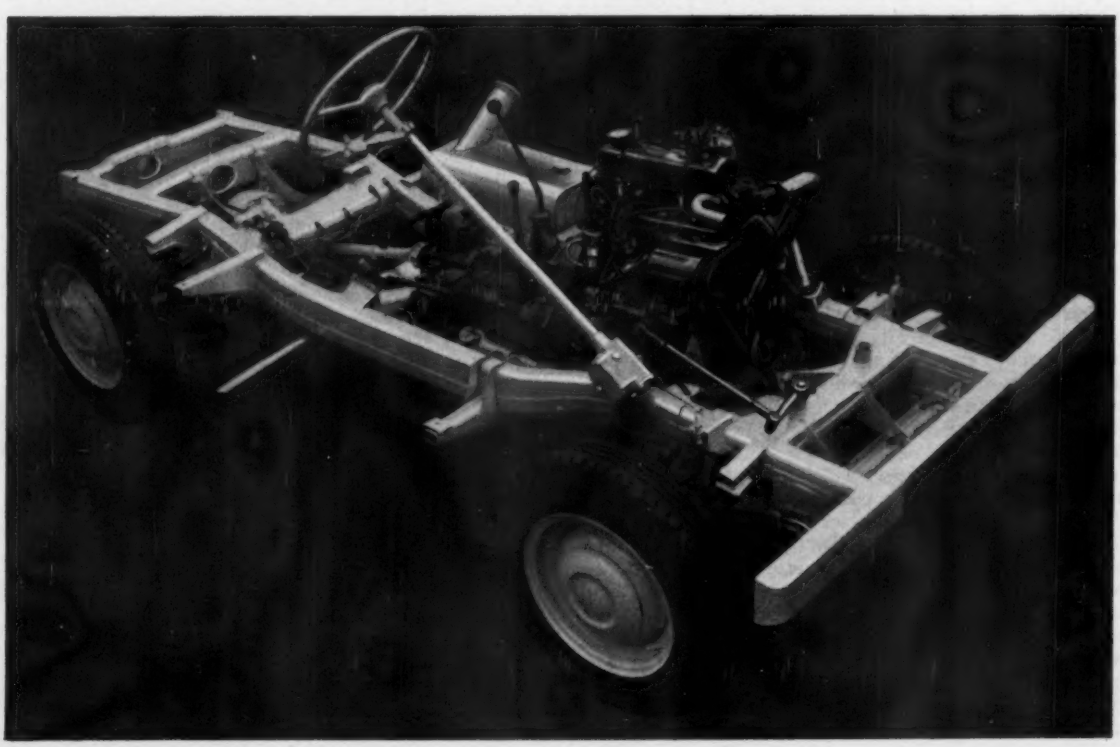
Both petrol and diesel engines have identical mountings, which are four circular bonded rubber biscuit type units, positioned wide apart approximately at the level of the axis of the crankshaft. Owing to the engine's being offset 2½ in to the left, in the chassis frame, the left-hand front mounting is slightly higher than the other to clear the frame sidemember. The forward mountings are attached to the front engine plate, and the rear ones each side of the transfer box.

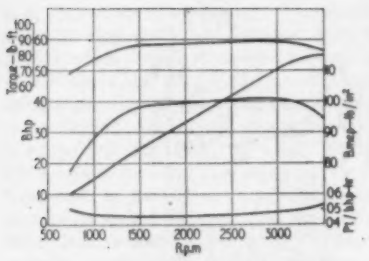
A conventional cooling system has been adopted. It is pressurized to the normal level of about 7 lb/in². Early models, fitted originally with a fan cowl, were found to overheat at high ambient temperatures. Removal of the cowling and the fitting of a slightly larger four-bladed fan cured this trouble. Nylon tubing, with standard olive type connections, is used for the fuel pipe line.

Transmission

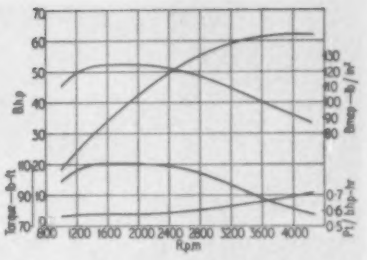
Some interesting features emerge from a study of the layout of the transmission line from the engine to the wheels. There is virtually no relative movement between the gearbox

The Austin Gipsy chassis has independent suspension on all four wheels, so there is virtually no relative movement between the gearbox and final drives





Power curves for the four-cylinder, 3½ in bore × 4 in stroke, 2,175 cm³ diesel engine. Good performance at high speeds is obtained with the Ricardo Comet III combustion chamber



Power curves for the four-cylinder, 3¼ in bore × 4¾ in stroke, 2,199 cm³ petrol engine, fitted with the Zenith 34 WIA carburettor, and an AC, tractor type, oil-bath air cleaner unit

and the final drives, but the shafts are considerably offset. Hardy Spicer 1310 series propeller shaft assemblies with needle roller bearing universal joints are used, and care is taken in the layout to ensure that the axes of the driving and driven shafts are set parallel so that a constant velocity transmission characteristic is obtained, irrespective of joint angularity.

Since the suspension system maintains the wheels constantly in one plane, the same type of Hardy Spicer assemblies can be used for the front and rear half-shafts, although it is admitted that on the locks some irregularity can be detected in the drive, owing to angular displacement of the front wheel stub shafts. This however, is not serious enough to warrant the use of constant velocity joints, which would involve total enclosure and other costly complications.

To clear the engine sump, the front final drive unit is offset 2¾ in to the right-hand side and, as previously mentioned, it is also offset ⅞ in forward of the axle centre-line, to obtain a larger lock angle. The rear final drive unit is installed mid-way between the wheels.

Unlike the Champ, the Gipsy has its transfer gearbox immediately behind the main gearbox. In the high transfer gear range the front wheel drive is automatically disengaged, and the drive to the rear axle is direct, the output shaft in the transfer box being co-axial with the gearbox mainshaft. Thus, although the gears are in constant mesh, they are relieved of any driving loads in high range. Engagement of low range simultaneously engages the front wheel drive. The ability of the driver to change from high to low or *vice versa* while the vehicle is operating under power is attributed to this direct drive high range feature. The change is possible, however, only if the vehicle is running straight at the time. No attempt is made in the design to compensate for wind-up of the front propeller shaft due to the greater aggregate distance travelled by the front wheels on the locks. It is felt—and this has been borne out in practice—that, as the front wheel drive is normally engaged only for negotiating soft ground, the wind-up is dissipated before it can build up to an undesirably high value.

Since the rear propeller shaft, which is very short, is connected to the transfer box at mainshaft level, the power unit is slightly inclined, to bring the transmission line towards that of the hypoid bevel pinion. A compromise is necessary, however, to avoid excessive displacement of the front propeller shaft. There is virtue in this compromise in that the universal joints are kept constantly in action, thus avoiding the danger, always present with inactive joints, of indentation of the journals and races by the needle rollers.

Clutch

Alternative sizes of the standard Borg and Beck single-dry-plate clutch are fitted—9 in diameter for the petrol engine and 10 in diameter for the diesel unit. Both sizes have a spring-centre plate, and they incorporate a sealed ball thrust release bearing. The release fork is secured by a cotter to a cross-shaft, on the projecting left-hand end of which is the operating lever, also cottered. The slave cylinder of the hydraulic remote control is mounted on the bellhousing and is directly connected to the lever by a push-rod with no adjustment.

Between the engine and the gearbox is a cast iron flywheel housing; the more normal Austin practice is to employ a deep bellhousing on the gearbox mounted directly on a rear engine plate. The reason for this departure is that a housing, being relatively cheap to produce in different forms for different applications, offers a more flexible expedient for production, although physically it is, of course, more rigid than the plate. Provision is made for ventilating and cooling

ENGINE DATA

	Petrol	Diesel
Bore	3¼ in	3¼ in
Stroke	4¾ in	4 in
Swept volume	2,199 cm³	2,178 cm³
B.h.p.	62 at 4,100 r.p.m.	55 at 3,500 r.p.m.
B.m.e.p.	125 lb/in² at 1,500 r.p.m.	101 lb/in² at 2,800 r.p.m.
Torque	110 lb-ft at 1,500 r.p.m.	89 lb-ft at 2,800 r.p.m.
Compression ratio	6·8 : 1	20 : 1
Minimum specific fuel consumption	0·58 pt/b.h.p.-hr	0·46 pt/b.h.p.-hr
Maximum drawbar pull	3,000 lb	3,000 lb

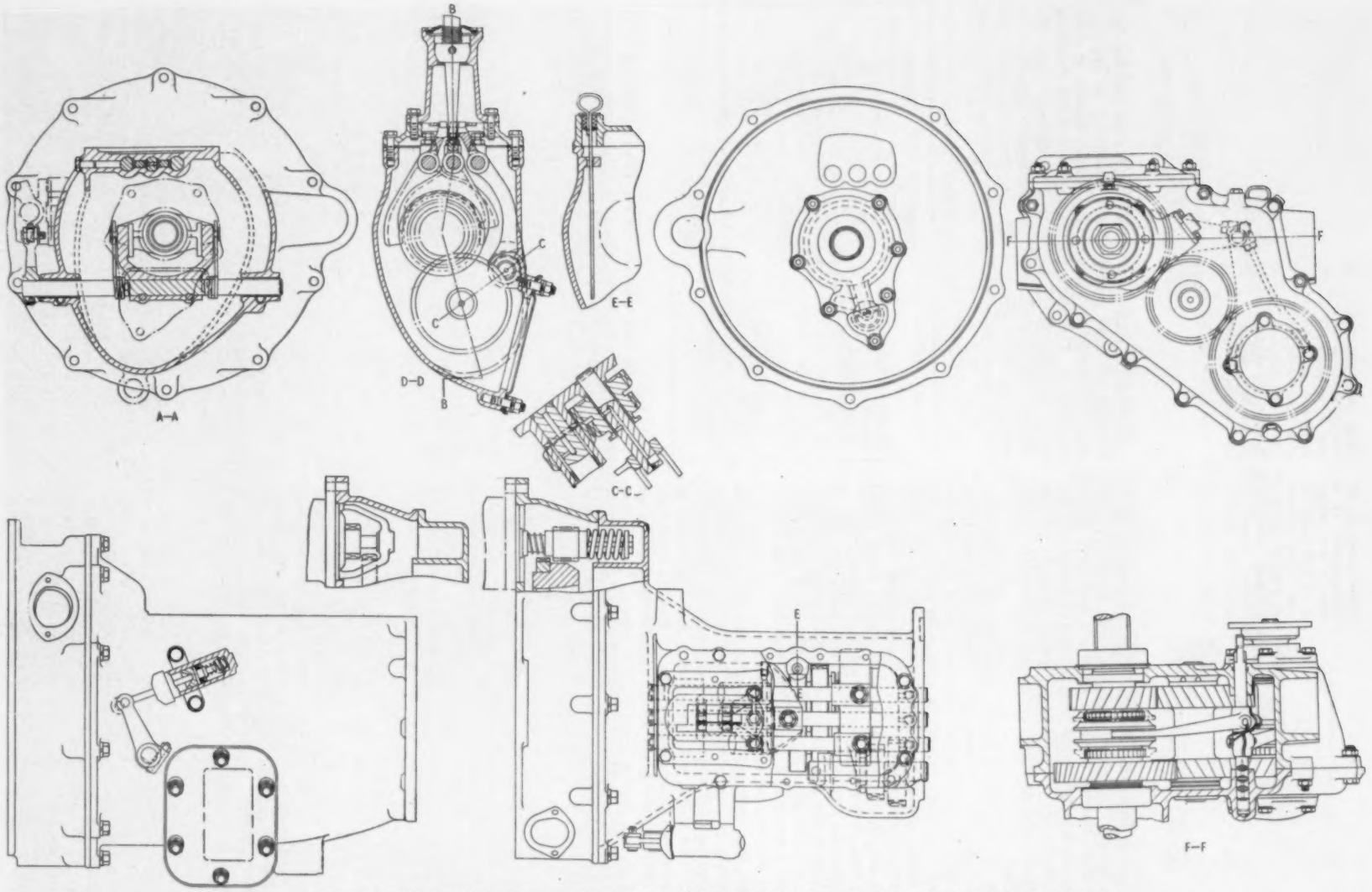
the clutch by two apertures in the flywheel housing, each having a cowled cover. The cowl of the upper opening is arranged to face forward and that of the lower opening rearward to induce a stream of air past the clutch.

Gearbox

Although entirely orthodox, the gearbox is a new design intended for application to the lighter range of commercial vehicles, the Gipsy being its first application. Four forward speeds and one reverse are provided, and there is synchromesh on top, third and second speeds. There are many features of the design that represent scaled-up car practice, and there is evidence of both Austin and Morris influence in the detail. The box is obviously intended for heavy duty, having shaft centres of 3⅛ in and a tooth width of ¾ in.

Die-cast in LM4-M aluminium alloy, the gearbox casing is integral with the shallow bellhousing. The layshaft is almost vertically below the mainshaft, and the selector rods are at the top. The cover carries only the lever turret. On the left-hand side is a standard S.A.E. regular duty power take-off mounting face, the drive being taken from the largest layshaft gear.

Stamped in En 36T, the primary shaft is supported in an Oilite spigot bush and in a ball bearing in the front wall of the gearbox. The outer race of the bearing is grooved and located by a spring ring retained by the front cover. This cover, which is retained by seven studs, is spigoted on the projecting register of the bearing, and is combined with the sleeve that carries the ball clutch release bearing. It also houses a lipped oil seal round the primary shaft, and locates the layshaft spindle. Standard S.A.E. 10 splines, 1¼ in diameter, are machined on the primary shaft, to receive the clutch plate, and the ball bearing is retained by a thin



VIEWS SHOWING THE GENERAL ARRANGEMENT AND SOME OF THE DETAILS OF THE GEARBOX AND TRANSFER BOX

Sections A-A and D-D are taken from the view on section B-B, which is illustrated on the opposite page

nut, locked by a tabwasher. The male synchromesh cone and the dog teeth are integral with the helical primary pinion which, being of much smaller diameter, is generated on a Fellows gear shaper and subsequently shaved. A 30 deg left-hand helix angle has been adopted.

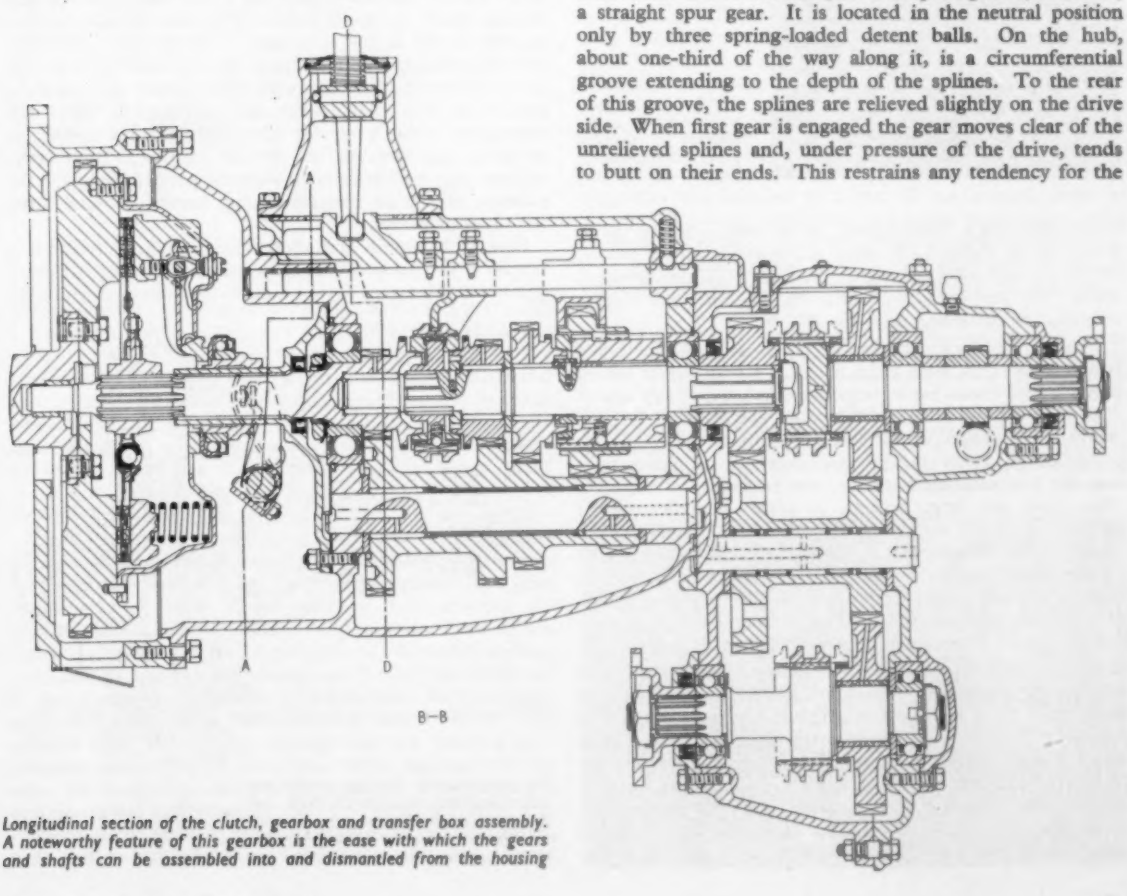
An En 36T stamping is also used for the mainshaft, which is supported at the rear by a ball bearing, and at the front by a spigot bearing consisting of 19 loose needle rollers. The shaft is ground to three diameters. It has three sets of straight-sided splines, and an integral flange between third and second speed gears. These two gears are in constant mesh, and are free-running on the shaft, each being carried on needle rollers. Identical rollers are used for both, there being 32 for the third speed gear and 33 for the slightly larger diameter of the second speed gear. The rollers run directly on the shaft and in the gears, and are lubricated through radial drillings in the gears. The third speed gear is located on the shaft by a splined collar, which is rotated, in a groove, out of register with the splines on the front end of the shaft, and retained in this position by a small spring-loaded plunger in the shaft. An integral synchromesh cone and large diameter dog teeth again necessitate the cutting of the gear on a Fellows shaper; the helix angle in this case is 33 deg 21 min 26 sec.

A helix angle of 30 deg is used on the second speed gear which, being of larger diameter than the integral dog teeth and cone, can be hobbed. There is evidence of long-standing Morris practice in the method of locating this gear. The splines for the sliding hub for first and second gear have a root diameter equal to the diameter of the second gear roller track, and the gear just passes over the splines. After the gear is placed in position, the needle rollers are inserted,

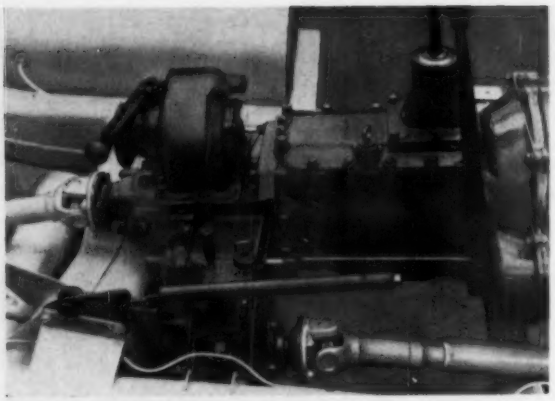
Steering adjustment is effected at the screwed-on ends of the connecting link interposed between the inverted drop arm and the relay lever

and the assembly is retained by a split thrust washer; this in turn is held in place in a recess in a splined collar, a tab on each half registering with a radial slot in the collar to prevent rotation. Like that of the third speed gear, the splined collar is turned out of register with the splines, but in this case a key is used to prevent axial movement of the thrust washer. A spring-loaded plunger locates the key, the projecting part of which is covered by the sliding hub.

Splined on the outside of the hub, which is free to slide a short distance on the shaft, the first speed gear is, of course, a straight spur gear. It is located in the neutral position only by three spring-loaded detent balls. On the hub, about one-third of the way along it, is a circumferential groove extending to the depth of the splines. To the rear of this groove, the splines are relieved slightly on the drive side. When first gear is engaged the gear moves clear of the unrelieved splines and, under pressure of the drive, tends to butt on their ends. This restrains any tendency for the



Longitudinal section of the clutch, gearbox and transfer box assembly. A noteworthy feature of this gearbox is the ease with which the gears and shafts can be assembled into and dismantled from the housing



View showing the installation of the full-torque power take-off, which can be driven with the main gearbox set in any speed

Rear view of the transmission installation and the frame cross-member cranked under the propellor shaft universal joint

gear to slide out of engagement. Since there is no first speed notch for the detent balls, there is not any tendency for the hub to move forward and engage the second gear synchromesh cones. All the gears, including the layshaft and reverse clusters, are machined from En 36T steel.

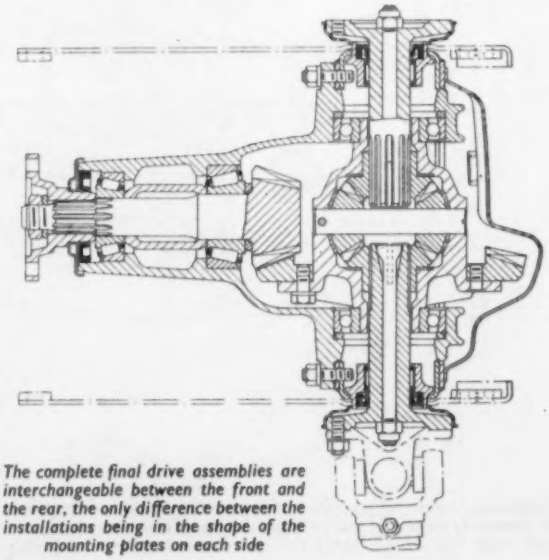
Synchromesh of the inertia lock type is used, the combined female cones and baulk rings being of a particularly robust pattern. All three baulk rings are identical. They are of B.S. 1400 AB1-C aluminium bronze, and have three lugs, which are V-shaped on the faces opposite to the open ends of the cones. Each ring fits in a recess in the sliding hub, and the lugs register in slots in the periphery of the hub; these slots are large enough to allow a predetermined amount of rotary movement. Inward projections on the internally toothed sleeve that slides on the hub are machined to match the lugs on the baulk ring; they are arranged so that the inclined faces of the two parts abut until the speeds of the cones synchronize, and then the sleeve is free to slide past the baulk ring lugs and engage the dog teeth.

All four gears on the layshaft are integral, forming a cluster which runs on needle roller bearings—33 needle rollers at each end—on a spindle machined from En 35 bar. The spindle is held in place between the front and rear covers, being prevented from turning by abutting shoulders on the front end and the cover. Hardened steel thrust washers are fitted at each end of the cluster gear, the rear washer being made in several thicknesses to regulate end-float on initial assembly. Oil troughs are formed in the castings at both ends of the layshaft; from them oil is fed through axial and radial drillings into the bearings.

The reverse cluster is normally at rest, and engages simultaneously with the layshaft and mainshaft first speed gears. It is bushed with Clevite or VE2 bronze, and the En 32B spindle is retained in the casing by a flat key fitting in a transverse slot on the end of the spindle and in a recess cut in the rear face of the casing. The front end of the spindle projects through a web in the casing, and has a large radial drilling, open end upwards, to collect oil, which passes through an axial drilling into two radial holes opposite the bushes in their operative position. At first, one hole was used, and it led the oil to the space between the two bushes. However, it was found that, no matter how copious the flow of oil, once it started flowing through one bush, the other tended to run dry. This trouble was overcome by drillings direct to each bush.

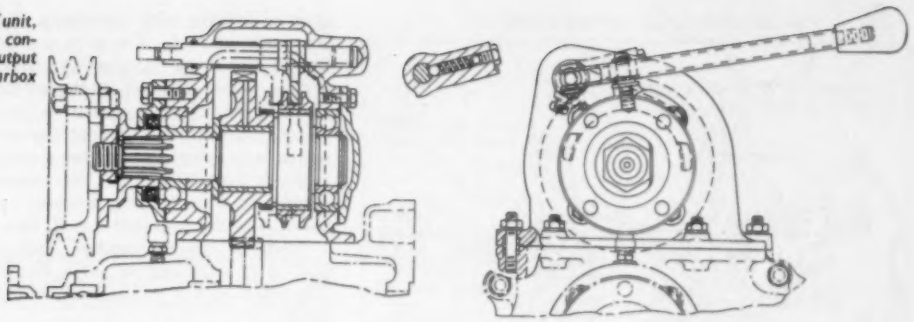
Aluminium bronze, B.S. 1400 AB1-C, die-cast selector forks are carried on the selector rods, which slide bodily in the casing. Since the gear lever is positioned well forward, the striker forks are separate from the selector forks. Both sets of forks are located on the rods by taper-ended setscrews. The cover fits over the top of the selector assembly with very little clearance between it and the setscrew heads: this obviates the possibility of their falling out into the box. Nevertheless, locknuts are fitted, together with Shakeproof washers. The rear ends of the rods are notched for the detent balls which, with their springs, are retained by the top cover. Plungers in a cross-drilling towards the front of the box provide the normal type of interlock. The lever turret is cast separately from the cover and, together with a guide plate, or gate, for the lower end of the lever, is bolted to it. A spherical dust-sealing cap covers the upper opening through which the lever passes. The inner member of this cap fits closely to the lever and is spring-loaded upwards, while the outer member is retained in the turret by a circlip. It was found that the diameter of the inner seal, and the size of the hole in the outer one necessary to allow full movement of the lever, did not allow sufficient overlap to ensure a seal; however, the trouble was cured completely, without any modification to existing parts, simply by the addition of a third member, fitting loosely between inner and outer members.

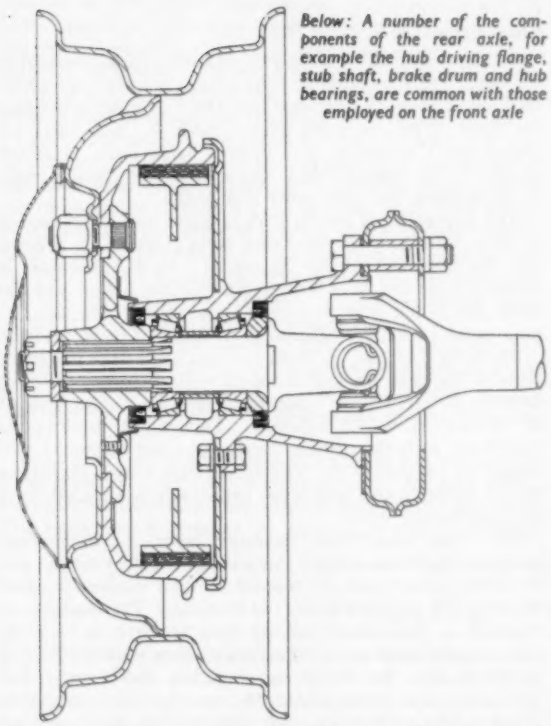
Ease of assembly and dismantling of the gearbox is a feature of the design. The mainshaft rear bearing is carried in a circular housing having a diameter slightly greater than



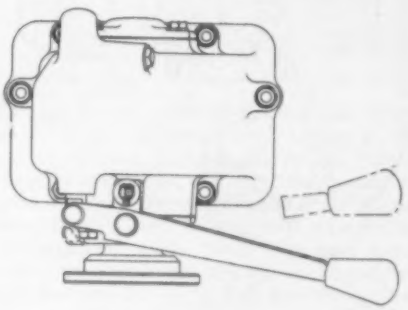
The complete final drive assemblies are interchangeable between the front and the rear, the only difference between the installations being in the shape of the mounting plates on each side

Right: In the power take-off unit, the free-running gear is in constant mesh with the helical output gear of the transfer gearbox





Below: A number of the components of the rear axle, for example the hub driving flange, stub shaft, brake drum and hub bearings, are common with those employed on the front axle



that of the first speed gear. After the layshaft cluster, with a dummy spindle to hold the needle rollers in place, has been dropped to the bottom of the casing, and the selector rods and forks removed from the top, the complete mainshaft and primary shaft assemblies can be drawn out through their respective ends of the box. This is evidence of the Austin influence in the design, as this feature has been incorporated in Austin gearboxes of different types for many years.

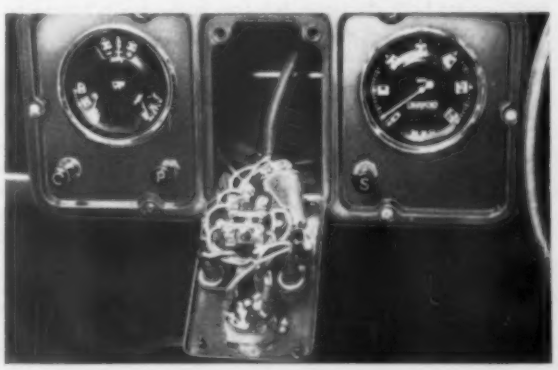
Transfer box

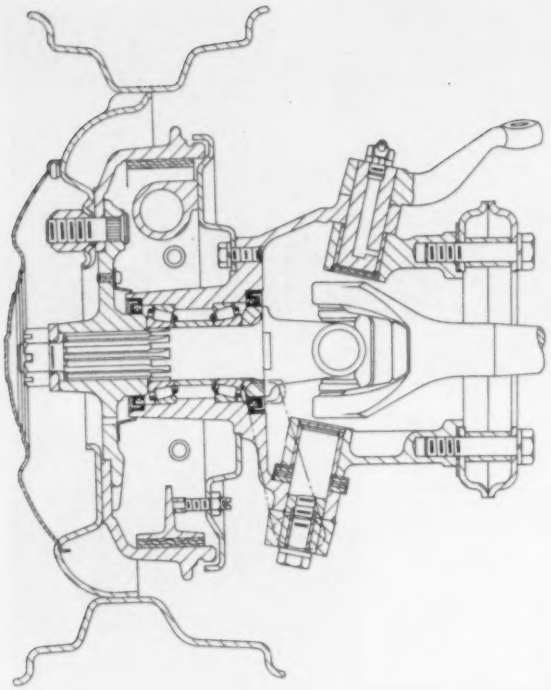
In effect, the transfer box is a simple two-speed auxiliary gearbox offering a reduction of 2·02 : 1 in all the main gearbox speeds, with coaxial input and output shafts and dog engagement of either direct drive or drive through the layshaft. An additional output shaft, for the front-wheel-drive, is driven by a gear meshing with the smaller layshaft gear, and has its own dog engagement. All three shafts are in line in a plane inclined at 45 deg towards the right-hand side, and the distance between their axes is 4½ in. The selector forks for the two sliding dogs are attached to a common selector rod, so that they move together. In this way, the front-wheel-drive is automatically coupled in low and uncoupled in high range.

Sand-cast in LM4-M aluminium alloy, the transfer casing is in two parts, the rear part being little more than a cover plate. The front part is attached to the rear face of the main gearbox by seven studs and two setscrews, the only register being the projecting outer race of the mainshaft ball bearing. The setscrews are inside the transfer casing and locked by tabwashers. All the nuts are outside, including one on a long stud which passes through the cover as well. The joint between the casing and cover is also flat, the two parts being located by two sleeve dowels. Eight bolts and four studs hold the two parts of the transfer box together. The layout of the box is such that it cannot be fitted to the gearbox as an assembly, but must be assembled piecemeal.

The input pinion is splined on the end of the gearbox mainshaft and retained by a nut. It has a tooth width of 1 in and a 30 deg right-hand helix, the same angle being used on the second train, which has a tooth width of ⅞ in. The pinion hub, facing towards the main gearbox, passes through a lipped oil seal with its lip facing the same way.

Removal of this panel, which is fixed simply by clips fitting in rubber grommets, discloses the lighting and ignition switch, direction indicator switch, flasher unit, and junction box with fuses and associated wiring. One of the panel lamps is detachable and can be used for illumination for work on this panel at night. After the panel has been pulled away, access can be gained through the hole that is uncovered, for the removal of the instruments on each side. This is one of the many examples of features giving evidence of the detail thought that has been devoted to the reduction of servicing costs and the simplification of maintenance





A two-piece king pin is employed, each piece being a shrink fit in one of the two steering levers

In the two gearboxes the oil levels differ so widely that it would be impossible to have a common reservoir of lubricant. The output gear is bushed, and runs freely on the output shaft, which carries a splined hub for the sliding dog at its forward end. Ball bearings carry the output shaft in an extension of the cover, which also houses the speedometer drive.

Four sets of caged needle rollers carry the layshaft cluster, on which the two gears are integral. The spindle is a tight fit in both the casing and the cover. It is located in the casing by a transverse flat key similar to that of the gearbox reverse spindle.

Identical with the rear drive output gear, the front drive gear runs freely on the lower output shaft, which is similar to the upper one, but reversed, except in that the bearings are at each end instead of having an overhung gear. The ball bearings are both the same, but that at the front, having no other protection, is furnished with a chip shield. The sliding dog and the selector fork are identical with those for direct drive.

Power take-off

In the top of the transfer box is an opening for a full-torque power take-off. This consists of a shaft with an integral splined hub carrying a sliding dog and a bushed free-running gear. The gear is in constant mesh with the output gear in the transfer box. A drive is therefore available for the power take-off in any of the main gearbox speeds whether the vehicle is in motion or not. The shaft is carried in ball bearings and projects through an oil seal to the rear, where it is splined to receive a standard driving flange for a Hardy Spicer joint or a direct-mounted pulley. The cast aluminium alloy housing is bolted to a flange on the transfer casing, but the standard S.A.E. face is not used.

There is also a standard belt-pulley, driven by bevel gearing in a housing bolted to the rear cross-member. It is connected to the power take-off by a long, Hardy Spicer shaft.

ELECTRICAL EQUIPMENT

Battery	12 volt, 51 amp-hr
Starter: petrol	Lucas M 418 G
diesel	Lucas M 45 G S1 D
Dynamo	Lucas C 39 PV2
Voltage regulator	Lucas RB 106/1

Final drives

Standard hypoid bevel carrier and differential assemblies are used, and the complete final drive assemblies are interchangeable between front and rear, the only difference being the shape of the mounting plates. Since the half-shafts are articulated, the casing, of truncated banjo form, is bolted rigidly to the chassis frame, and special provision is made for supporting the short inner stub shafts.

The hypoid bevel pinion shaft is offset $\frac{7}{8}$ in. It is carried in taper roller bearings, the outer races of which are pressed into the front and rear of the carrier. Mesh adjustment is effected by means of a washer between the pinion and the inner race of the larger bearing. These washers are supplied in a range of eight thicknesses, in steps of 0·002 in from 0·208 in to 0·222 in. A distance-piece separates the inner races of the bearings, and shims in varying thicknesses from 0·004 in to 0·030 in are inserted to give the correct preload of 16-18 lb-in to the bearings. This whole assembly is pulled up, with the driving flange, by a nut tightened to a torque of 135-140 lb-ft. The flange hub passes through a lipped oil seal, and additional protection is afforded by a dust cap on the flange.

Ball type semi-thrust bearings carry the differential assembly in split housings. Adjustment for the bearings and for crown wheel mesh is provided by thick washers inserted between the outer races and the housings. The washers are supplied in thicknesses varying from 0·175 in to 0·193 in, and are selected in pairs. Since the washers must be inserted sideways, with the differential assembly, this arrangement does not provide for preloading the bearings to any significant extent. The crown wheel is spigoted on the one-piece differential cage and retained by 10 setscrews, locked in pairs by locking plates. Both differential pinions run directly on the spindle, which is retained in the cage by a locking pin; spherical bronze thrust washers are interposed between the pinions and the cage. The side bevel gears run directly in the cage, and bear against flat, bronze thrust washers. Both the crown wheel and the pinion are machined from En 36T steel.

Splined into the differential bevel gears are the short stub axle shafts. These shafts are formed integrally with the driving flanges, and are supported at their outer ends in caged needle roller bearings, which run in races welded to the truncated banjo casing. The problem of fixing the stub shafts in the differential, arising from the fact that the one-piece cage denies access to the inner ends of the shafts, has been overcome in a simple way: the shafts are hollow, their bores being tapered at their inner ends, which are split; spherical headed expander bolts are inserted and after assembly of the shafts are tightened from the outer ends to a torque of 60-70 lb-ft. Self-locking nuts of the nylon insert type provide an oil seal for the threads. To reduce to a minimum the overhang loads on the needle roller bearings, the flanges are kept as close as possible to the casing, and for this reason, studs are used for the attachment of the Hardy Spicer joints. Alignment of the carrier

assembly with the stub shafts is, of course, extremely important, and shims are supplied for the carrier flange joint. These are in coloured plastics in thicknesses of 0·002 in (blue), 0·003 in (green) and 0·006 in (clear). The mounting plates are attached to the casing by four $\frac{5}{16}$ in diameter studs on each side. They are of 10 S.W.G. plate, and are reinforced at the points of attachment to the chassis frame by welded-on bushes.

Wheel assemblies

The front and rear hub assemblies, although not identical, have a number of common parts, including the hub driving flanges, stub shafts, brake drums and hub bearings. Malleable castings to B.S. 310-2 specification are employed for the hubs themselves. The outer races of the taper roller bearings are pressed into the hubs from each end, where lipped oil seals are also fitted. The outer section of the driving shaft, which is integral with the yoke of the outer Hardy Spicer joint, passes through the bearings and is splined on to 10 parallel splines, $1\frac{1}{4}$ in diameter, on the driving flange. This flange is pulled up to the bearings by a castellated nut tightened to a torque of 120-140 lb-ft. The inner races of the bearings are separated by a distance-piece, and shims, 0·002, 0·003, 0·010 and 0·030 in thick, are used for adjustment to give slight end-play. In each case, the inner oil seal bears on a collar behind the inner race of the inner bearing. The five wheel-studs are riveted to the driving flange, those on the left-hand side having left-hand threads.

At their inner ends, the rear hubs are bell shaped, and formed on each is a flange and spigot for the bolted attachment to the suspension arm. Bolts are also used for the attachment of the rear brake back-plates to flanges on the hubs.

Each front hub is opened out at its inner end to form the outer member of an open swivel assembly. Because of this arrangement the brake back-plate is attached by setscrews. The inner swivel member, or support, is also a malleable casting with the inner face formed into a flange and spigot for the attachment to the suspension arm. Owing to the shape of the swivel support, setscrews driven into tapped holes in the casting are used for attachment of the suspension arm. The support member is furnished with Clevite bushes for the upper and lower king pins, the inner ends of the bushes being sealed by expansion plugs. Both king pins are a shrink fit in the upper and lower steering arms, and are threaded through bores in the swivel hub into the bushes. The steering arms are attached to the swivel hub by two setscrews with sleeve dowels to take the shear stresses. On the side opposite to that on which the steering gear is mounted, where there is no upper steering arm, a dummy flange is used to retain the upper king pin. The weight of the front of the vehicle is taken on steel and bronze thrust washers on the lower king pins, between the swivel hub and support. Lock stops are provided by setscrews driven into lugs on the swivel hubs, washers being used for adjustment: the heads make contact with bosses on the supports.

TRANSMISSION SPECIFICATION

Clutch:		
petrol	Borg and Beck 9 in diameter	
diesel	Borg and Beck 10 in diameter	
Auxiliary reduction	2·02 : 1	
Final drive tooth ratio	8 : 41	
Hypoid offset	$\frac{7}{8}$ in	
Gear ratios:		
top	1 : 1	
third	1·37 : 1	
second	2·35 : 1	
first	4·05 : 1	
reverse	5·168 : 1	
Overall ratios:	high	low
top	5·125 : 1	10·352 : 1
third	7·021 : 1	14·182 : 1
second	12·043 : 1	24·326 : 1
first	20·756 : 1	41·927 : 1
reverse	26·450 : 1	53·400 : 1

STELLITE PISTON RINGS

HEPWORTH and Grandage Ltd. are now able to offer piston and sealing rings produced entirely from Stellite material, to work efficiently under conditions that would prohibit the use of all the previously accepted piston ring materials.

Stellite, which, of course, is already widely used for valve and valve seat facings has the following outstanding properties:

(1) Excellent resistance to collapse at high temperatures—it will operate satisfactorily at temperatures of 1,000 deg F (540 deg C).
(2) It is particularly resistant to wear and can be run successfully in bores of Stellite, hardened steel, hard chromium plate, most bronzes and cast irons, provided no abrasive particles are present. It is not advisable to run Stellite against mild steel or soft stainless steel, as it is liable to gall.
(3) Corrosion and erosion resistance—it can be used with confidence in many applications where ferrous rings would be unsuitable.
(4) High modulus of elasticity and tensile strength, and the rings can be designed to exert very high wall pressures, if required.

Naturally, Stellite rings are expensive, but there are applications where the disadvantage of the high initial cost is far outweighed by subsequent economies in maintenance costs and loss of productive time while changing or renewing rings. Typical applications are: high temperature steam valve seals, pressure die-casting machine plunger rings, pumps and compressors handling corrosive fluids, and any application calling for an exceedingly tough, heat-resistant, wear-resistant material.

Although Stellite is available in many grades, the Stellite Grade 6 has, so far, been found to be the most adaptable and suitable for piston rings. Being cobalt-based and alloyed with chromium and tungster, it has the following physical properties: ultimate tensile strength 46 ton/in², modulus of elasticity, or En. value, 30×10^6 lb/in², hardness, as cast 350-425 D.P.N. Further information can be obtained from Hepworth and Grandage Ltd., Bradford 4.

The Autocar

OUR associated journal, *The Autocar*, is producing three special issues to give full coverage to the private car Show at Earls Court. The first is the London Show Guide, and it will contain, among other things, floor plans showing the arrangement of the stands, and a guide to outstanding exhibits. In the next issue, the London Show Report, which is treble-size, there will be a stand-by-stand survey, including descriptions, specifications and illustrations. The third issue is the London Show Review, which will be a critical appraisal of 1959 design trends. These issues, which will be published on the 17th, 24th and 31st October respectively, can be obtained from any newsagent; the price is 1s, except for the London Show Report, which is 2s 6d.

ENGINE DYNAMOMETER

English Electric Equipment at the Laboratories of Esso Research Ltd.

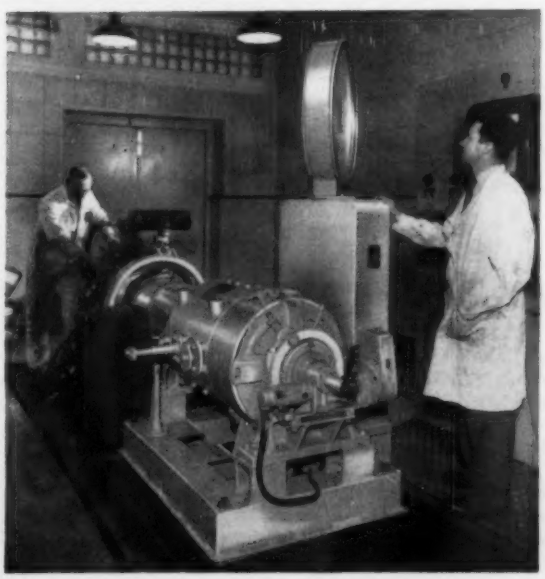
As world-wide demand for fuels and lubricants for internal combustion engines continues to increase, the oil industry has had to evolve new refining techniques to meet it. As a result, the quality of these products has been steadily improved, and this has contributed in no small measure to the advances in power, fuel economy and long life, which are the goal of every manufacturer and the desire of every consumer. However, these advantages, and especially improved fuel economy, can only be gained from more efficient use of fuel.

A gallon of petrol should yield about 100 million foot-pounds of work; but for thermodynamic reasons, only a part of this tremendous potential energy can be converted into useful work by the automobile engine. A thermal efficiency of about 30 to 35 per cent is all that can be expected for petrol engines; diesel engines are slightly more efficient, at about 40 to 45 per cent. These figures are further reduced by friction in bearings, between pistons and cylinder walls, and in other moving parts.

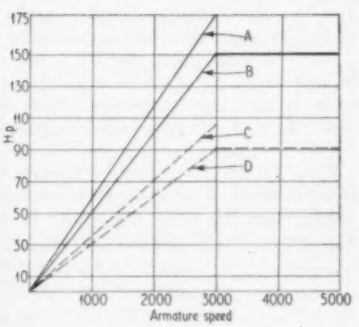
Much can be done to minimize these losses, without at the same time impairing the operating life of the engine. Good engine design can reduce the power required, for example, to pump air into the combustion chamber and to remove the exhaust gases. Also, the petroleum research scientist, by developing lubricants of lower viscosity but still with adequate lubricating properties, can decrease the viscous drag. The product quality, of fuels as well as of lubricants, can only be improved by operational experience. For this and other reasons, every progressive oil company is continually testing fuels and lubricants in representative full-scale equipment.

To accommodate the expanding scope of the work of Esso Research Ltd, at their laboratories near Abingdon, Berkshire, new buildings of advanced design have recently been completed. The construction of the new engine test laboratory included in this expansion scheme provided an opportunity of installing up-to-date equipment of the latest design. Among the test cubicles in the new engine laboratory is one equipped with an English Electric dynamometer; this is already in use for measuring output horsepower and for motoring engines under test, Fig. 1.

Fig. 1. The English Electric swinging frame electric dynamometer



This equipment consists of the swinging-frame dynamometer; an A.C. mains-connected motor-generator set, for absorbing the electrical power output of the dynamometer and supplying a variable D.C. voltage for motoring the dynamometer; and interlocked and protected push-button controls for the complete electrical circuit, as shown in Fig. 2.



A *1 hour absorption rating;* **B** *continuous absorption rating;* **C** *1 hour motoring rating;* **D** *continuous motoring rating*

Fig 3. Curves showing the characteristics of the engine dynamometer at Abingdon

A D.C. dynamometer is employed. It has been developed from an English Electric traction motor, with an armature suitably constructed for high-speed operation. The machine frame is mounted on trunnion bearings to allow free rotation through a limited arc; one end of a torque arm is solidly connected to the machine frame and the other end to an Avery dial-reading and gravity-controlled weighing gear. Thus, the horsepower transmitted by the armature shaft can be calculated from the load on the torque arm. A knife-edge is mounted on the machine frame, opposite the torque arm, so that the weighing gear can be calibrated by the direct addition of weights.

The two-fold duty of the machine is absorption and motoring. As a load for the test engine, the dynamometer is capable of absorbing up to 150 h.p. at a maximum speed of 5,000 r.p.m. or, alternatively, 175 h.p. at 3,000 r.p.m. for one hour. When motoring to drive an engine, for example, for hot friction tests, the dynamometer and D.C. supply system are capable of operating at about 60 per cent of the absorption ratings. Fig. 3 shows the absorption and motoring characteristics for the continuous maximum and one-hour ratings.

For absorbtion, the motor-generator set, Fig. 4, provides the electrical load; and for the motoring tests it supplies variable voltage to the dynamometer. It consists of an A.C. slip-ring induction motor-generator, a D.C. generator-motor and a constant voltage exciter, all directly coupled together on a common bedplate. When the dynamometer is required for motoring duty, the A.C. motor drives the D.C. machine

as a generator, whose output can be governed manually by excitation control. In this way, a variable voltage D.C. supply is provided for the dynamometer. When the dynamometer is acting as a load for the test engine, the dynamometer output is fed into the D.C. machine, which then operates as a motor, driving the A.C. machine as an induction generator. The A.C. output of the induction machine is fed back into the A.C. supply system, which thus absorbs the energy produced by the test engine.

The dynamometer incorporates a number of mechanical design features that reduce torque losses due to restriction of frame movement. For example, the electrical connections to the machine are flexible leads brought out from the underside of the frame. Similarly, precautions have been taken to ensure that the forced ventilation air ducting for the machine does not impede the rotational movement of the frame; cold air from outside the building is blown through a sheet steel duct by a fan beneath the test cell floor, and goes through a clearance entry into the air circuit of the machine. To prevent vibration damage to the trunnion bearings, the outer races are rotated manually at regular intervals.

A control desk is situated in the test cubicle. The controls include push-buttons marked 'control on', 'main on', 'control/main off', 'increase load/decrease speed', 'decrease load/increase speed', and 'emergency stop'. Also provided on the control panel are ammeters and voltmeters, and indicating lamps for 'main circuit closed', 'paralleling-fault', 'control available' and so on. Accurate measurement of fuel consumption is effected by using a fuel weighing device developed by Esso Research. The pressing of a single button results in the measurement of both the rate of flow of fuel and engine speed.

The control marked 'paralleling-fault' is incorporated for the following reason. During the starting procedure, it is essential for the voltage across the terminals of the generator of the motor-generator set to be identical in magnitude and direction to the voltage across the terminals of the dynamometer before the main D.C. loop-circuit is closed. The closing of the D.C. loop during the starting operation is termed paralleling the two D.C. machines. To prevent the circuit from paralleling while the voltages are unbalanced, the relay has been introduced across the terminals of the D.C. loop closure contactor. Should an attempt be made to parallel the two machines, while the voltages are unbalanced, this relay initiates interlocks in the control circuit which, in turn, trip the D.C. loop contactor, and switch on a lamp on the main control desk. This lamp indicates that the paralleling relay needs resetting.

A reduction bevel gear, driven by the dynamometer armature shaft, actuates an electric tachometer. It also drives a centrifugal over-speed device that can be set to cut off the ignition and fuel supply to the test engine in the event of dangerous over-speeding. Other relays and control circuits operate to protect the auxiliary equipment of both the dynamometer and test engine.

The main purpose of the dynamometer in this installation is the measurement of power loss, air pumping and viscous drag in internal combustion engines. Although more accurate methods of determining friction losses are known, for example, the electronic integration of power output curves, the more commonly used types of torque measuring equipment are accurate enough for this particular application. In addition to providing sensitive measurement of engine torque over a wide range of power and speed, the swinging-frame electrical dynamometer has other important advantages. It can be used as a motor for driving the engine, and can be changed over instantaneously from absorption to load duty and back again, as required. This latter feature enables the research engineer to measure the friction and pumping losses of the test engine, under any conditions of engine operation, before the engine cools off or conditions change in any way. The dynamometer can be used to provide the motoring power so that these losses can be accurately analysed, for example, when experimental lubricants are being tested, by successively removing the various engine components, that is, the head, piston rings, and so on, down to the bare crankshaft.

Fig. 2. Above: The control desk for the engine dynamometer at Esso Research Ltd.

Fig. 4. Right: The motor-generator set for providing the electrical load or supplying a variable voltage to the dynamometer

Apparatus of this type assists the petroleum technologist in keeping in step with changes introduced by the automobile engineer. He can quickly determine quantitatively the improvements in performance due to fuel and lubricant quality changes, and thus he continues his search for future improvements in efficiency. This will be an important factor in maintaining the competitive position of the internal combustion engine in both the home and overseas markets.

SHELL FILMS

THE Shell Petroleum Co. Ltd. have recently introduced a new film catalogue, which is available to representatives of organizations. This catalogue lists about 166 films on an extremely wide range of subjects. Among the subjects are physics, the internal combustion engine, automobiles and motor cycles, and the history of science and technology. There are also films on oil production, refining, transport, products, and management and personnel. Further details can be obtained trom the Film Section, Public Relations Department, Shell Petroleum Co. Ltd., 1 Kingsway, W.C.2.

PISTON RING WEAR

*The Application of Radiotracers to the Measurement of Engine Wear in Moving Vehicles**

J. H. DETERDING, M.A.† and J. R. B. CALOW, B.Sc.†

A SMALL amount of wear of the piston rings and cylinder bores will drastically impair the efficiency of an internal combustion engine and it is generally this factor which sets a limit to the length of time before an overhaul becomes necessary. At the same time, the high temperature and corrosive atmosphere, produced by the combustion reaction, subject the lubricant in this region to the most severe conditions to be found in the engine. It is not surprising, therefore, that considerable attention has been given for many years to the problem of piston ring and cylinder bore wear.

Conventional methods of studying ring and bore wear, by the measurement of the change in weight or dimensions of the components over a period of operation, have yielded valuable information, but suffer from two main disadvantages. Relatively long runs are necessary to produce significant results and, more important, serious variations are introduced into the results by the need to dismantle the engine, clean and measure the parts and re-assemble between runs.

The availability of radioactive isotopes in recent years has led to the development of an alternative method, which overcomes these difficulties. By using an irradiated piston ring, wear of this component can be detected by the presence of radioactive wear products in the oil. If suitable instruments are employed, continuous and automatic records of ring wear can be obtained. The sensitivity of detection of the radioactive wear debris is high and enables comparisons of ring wear under various conditions to be made in a very short time. The need for dismantling the engine for measurement is, of course, eliminated.

This technique is being used successfully by many workers for the continuous measurement of piston ring wear in static engines on the test bed. The results of work in a laboratory engine, however, must be applied with caution to engines operating in the field, since the conditions under which the laboratory engine operates are not necessarily identical with those of an engine in a vehicle on the road. For example, the intake of airborne abrasive dust particles by an engine is probably higher on the road than in the laboratory. Again, the time taken for an engine to warm up after starting from cold could be different under the two circumstances and might lead to serious differences in the amount of wear caused by the action of corrosive acids during the warming-up period.

Several workers have extended the radiotracer method to the measurement of piston ring wear in cars on the road, by monitoring the sump oil before and after a period of running. After preliminary investigations with this technique in a Morris Minor car, it was concluded that, whilst useful results could be obtained by this means, the method was not being used to its fullest advantage. It appeared that the study of short-lived wear phenomena, such as that occurring on starting the car, could be made more effectively and in greater detail by the continuous measurement of the radioactivity in the oil.

As a result of considerations such as these, it was decided to apply the radio-isotope technique to the continuous measurement of ring wear in a vehicle. The main problems associated with taking this step were in the design and development of electronic and ancillary equipment to operate on the limited power supply available in the vehicle. It was also necessary that the instruments should be compact, and robust enough to withstand the vibration and sudden accelerations and decelerations that are unavoidable in a moving car.

This paper describes the apparatus that has been developed and fitted into a Standard Vanguard estate car to measure and record wear whilst the car is in motion. Its main use to date has been for research into lubricating oils. Some typical results are also discussed.

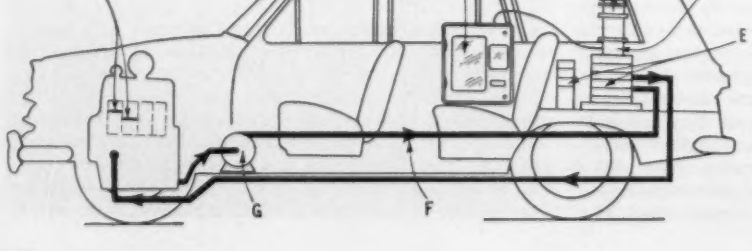
Irradiation of the piston rings

Piston rings of the same composition as those normally fitted into the car are sent to the Atomic Energy Research Establishment, at Harwell, for irradiation. They remain in the pile for 12 weeks in a neutron flux of about 8×10^{11} neutrons/cm²/sec and on withdrawal contain the isotopes shown in the Table.

After the rings are removed from the pile, they are retained at Harwell for a few days to allow the radiation from the short-lived manganese isotope to fall to a safe level. They are then packed in lead containers and transported by rail. Four weeks after irradiation, the total gamma activity of the ring is about 4 millicuries, the principal sources of gamma radiation being the isotopes ^{59}Fe and ^{60}Co. In the work under discussion, gamma activity in the oil is measured, in preference to beta activity, as it has been found in earlier

*A Paper presented to the United Nations Conference on Atomic Energy, Geneva, 1st—13th September, 1958.

†Shell Research Ltd., Thornton Research Centre, Chester.



A radioactive piston rings; B counting and recording unit; C cooling fan; D scintillation counter head; E lead shielding; F oil flow; G pump unit

Fig. 1. Diagram illustrating the layout of the wear recording equipment in the vehicle

work[2] that the former bears the more satisfactory relationship to ring wear.

Measurements have been made of the decay of gamma activity in a solution containing a small piece of an irradiated ring which had been dissolved in acid. It has been found that the observed decay curve is consistent with that to be expected from a mixture of ^{59}Fe and ^{60}Co in the proportions indicated in the Table. Four weeks after irradiation the effective half life is about 50 days.

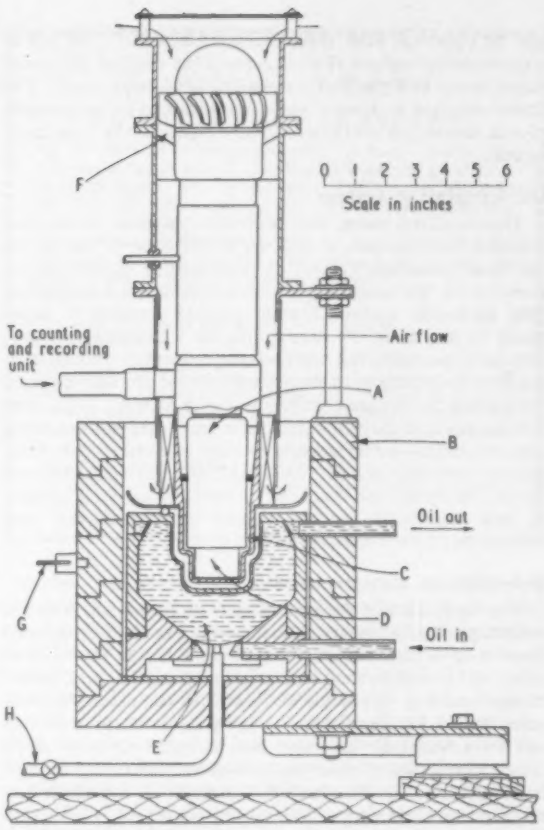
General description

A Standard Vanguard estate car was used for this work. It is powered by a four-cylinder water cooled petrol engine, which develops 68 b.h.p. at 4,200 r.p.m. This type of car was chosen because it has a relatively long body with a readily accessible and roomy luggage compartment situated behind the rear passenger's seat, Fig. 1. These features were considered desirable so that the radiation detector, with its bulky lead shielding, could be mounted as far as possible from the radioactive rings in the engine. Subsequently it has been found that the equipment can be used satisfactorily in a much smaller car.

The arrangement of the apparatus in the car is shown in Fig. 1. Two irradiated top compression rings are fitted on numbers 1 and 2 pistons, that is, the front two. The lubricating oil carrying the radioactive wear products, is drawn from the sump by means of an auxiliary pump unit powered from the car's 12 volt electrical system. This oil is then circulated through suitable piping to a counting cell at the rear of the car and thence back to the sump. The volume of oil in the the cell is about 800 ml.

Wear products in the oil in the counting cell are detected by a scintillation counter head, which contains a sodium iodide crystal and photomultiplier tube. The cell and detector are surrounded by a lead castle and, in addition, lead blocks are interposed between the castle and the engine, to provide better shielding from direct radiation from the active rings. The crystal housing projects into a recess in the cell and is kept cool, despite the presence of hot oil, by air from a fan mounted above the detector. Scintillations in the crystal, caused by gamma emission from the wear debris, produce electrical pulses in the photomultiplier. The pulse rate is proportional to the concentration of active material in the oil and therefore to the accumulated ring wear.

Pulses from the photomultiplier are fed to a counting and recording unit, which is on the rear seat on the car. After suitable amplification and scaling down in this unit, the pulses move a stylus in small steps across the chart of the clockwork-driven recorder. It is arranged that the stylus moves across the chart for three minutes and then returns to its zero position. The display on the chart is therefore in the form of a bar diagram with the length of the lines representing successive three-minute counts of the radioactivity in the oil. As the oil is continuously circulated to the detector, wear of the rings is indicated by the increase in length of successive lines on the chart and the wear rate



A photomultiplier; B lead shielding; C silicone fluid; D NaI crystal; E oil tank; F cooling fan; G ^{60}Co test probe; H drain

Fig. 2. General arrangement of the scintillation counter

TABLE : Radioactive constituents of an irradiated piston ring

Element	Relative amount in ring per cent weight approx.	Radioactive isotope	Half-life[1]	Radiation[1]	γ activity of ring after 12 weeks' irradiation at 8×10^{11} n/cm^2/sec and 4 weeks decay. μC/gm approx.
Iron	96	^{55}Fe	2·94 yr	Electron capture	—
		^{59}Fe	45·1 days	$\beta + \gamma$	267
Cobalt	0·01	^{60}Co	5·25 yr	$\beta + \gamma$	40
Chromium	0·34	^{51}Cr	27·8 days	Electron capture $+ \gamma$ (~8 per cent)	14
Phosphorus	0·64	^{32}P	14·3 days	β	—
Silicon	2·05	^{31}Si	2·65 hr	$\beta + \gamma$ 0·07 per cent	—
Manganese	0·85	^{56}Mn	2.58 hr	$\beta + \gamma$	—

can be observed over three-minute intervals. The use of transistors throughout this instrument has enabled the power requirement and physical dimensions to be kept small. The power required is about 6 watts; it is supplied by an auxiliary 12 volt battery, to avoid interference from the car's electrical system.

Oil circulation system

The auxiliary pump unit which is designed to circulate the oil to the detector, is situated on the floor of the car, by the front passenger's feet. It comprises a 12 volt motor, powered by the main car battery, which drives a swashplate type hydraulic pump. During normal running it draws about 12 amp from the battery; the car's charging system is able to cope with this current requirement. Provision is made for switching to an alternative battery, should prolonged circulation be required while the car is at rest. The total volume of oil in the whole system is ten pints, and the pump unit can circulate this volume through the cell in the three-minute counting period. Designed to withstand pressures up to 300 lb/in², the cell itself is made of steel. Pressures of this magnitude are experienced when pumping very viscous oils at low temperatures.

Scintillation counter head

A cross section of the scintillation counter head, with the counting cell and lead castle, is shown in Fig. 2. The detector consists of a thallium activated sodium iodide crystal, 1 in long × 1½ in diameter, and a photomultiplier in a finned copper housing. Since the oil in the cell can reach temperatures above 100 deg C, the housing is insulated from the cell by a 5 mm air gap, and heat is removed from it by conduction to the copper fins, which are cooled by air flow from the fan. In this way the detector is maintained at about 3 deg C above the ambient temperature. A resistance thermometer indicates the crystal housing temperature, on a meter in the counting and recording unit.

To keep the distance between the crystal and the oil to a minimum, the crystal was inserted directly into the copper housing which has a reflecting chromium plated inside surface. The photomultiplier is maintained in position by spring-loading. High voltage supplies are provided through multi-cored cable, which also carries the output pulses to the counting and recording unit.

Counting and recording unit

A single unit contains a strip-chart recorder and the electronic components for providing the voltage supplies for the photomultiplier and counting the output pulses. It has been especially designed to operate from a 12 volt battery and makes extensive use of transistors. The high voltage supplies for the photomultiplier are obtained from a converter of the ringing choke type, using a single transistor, which can provide up to 2,000 volts at low power. Stabilization of the output voltage is effected by a corona discharge tube, and doubling circuits are used to provide the various voltages for the dynodes of the photomultiplier[3].

Output pulses from the photomultiplier are fed into an impedance converter and amplifier containing two transistors, which amplify and shape the pulses suitably for counting. The amplifier operates in such a way that the change in count rate is less than 0·25 per cent for a 10 per cent change in discriminator voltage, and less than 0·01 per cent per volt change in the 1,550 volts supply to the photomultiplier.

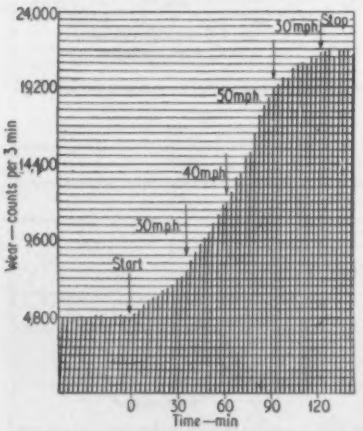


Fig. 3. Diagram showing the effect of speed on piston ring wear

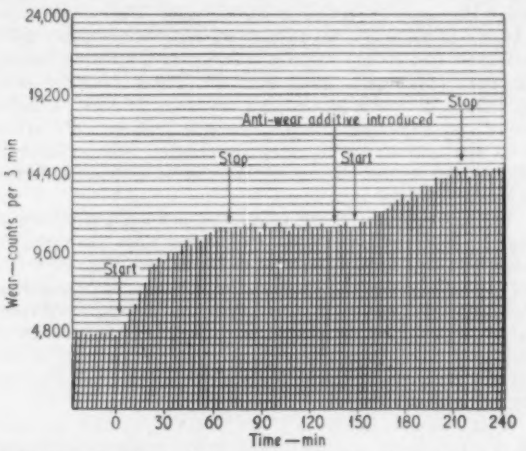


Fig. 4. Variation of rate of piston ring wear after the engine has been started from cold, and the effect of anti-wear additive on that rate

The output from the amplifier is fed to a combination of proprietary transistorized binary and decade units, which also act as pulse height discriminators. Switches are incorporated for the selection of various scaling-down ratios from 1 to 1,600. The resulting output pulses are then applied to a trigger circuit, which operates a relay for mechanical recording of the pulses.

A strip chart mechanism, with pressure sensitive paper marked by a stylus, is employed in the recorder. For every pulse applied to the trigger circuit, the stylus moves 0·02 in across the paper. A continuous line is thus drawn across the paper at a rate determined by the photomultiplier output and the selected scaling-down ratio.

After a counting time of three minutes, the stylus is lifted from the paper and returned to its zero position. This timing is controlled by the clockwork motor that drives the chart paper. There is a fixed interval of five seconds between successive three-minute counts.

Sensitivity

When the scintillation counter is assembled in the vehicle, the crystal is shielded by 1½ in of lead, and the addition of lead blocks provides a further 4 in of shielding against direct radiation from the radioactive piston rings. The background count rate, with fresh oil in the system, is then about 7 per sec. Efficiency of the detection of gamma-radiation, from a volume of 800 cm³ of liquid in the cell surrounding the crystal, is about 4 per cent. This was determined by the use of a solution containing a few milligrams of material from a radioactive ring.

If rings with a specific activity as given in the Table are used, and the total volume of oil in the lubricating system is 10 pt, there is an increase in count rate of about 100 per

second per milligram of wear. Thus, in a counting time of three minutes, a significant increase in count can be obtained from 10^{-5} gm of wear, which is about fifty times less than can be determined reliably by measurements of ring weight loss.

Safety considerations

When a four weeks' decay period has elapsed, the rings are fitted into the engine with the aid of specially designed long-handled tools. The rings are unshielded for only a few minutes during installation, when eyeshields and gloves are worn to give protection from the beta radiation.

After assembly, the radioactive rings are shielded by the water jacket, cylinder head, etc., so the radiation level outside the car is very low. The level at the front and sides of the car is less than 1½ millirads/hr, and on top of the bonnet, directly above the rings, it is about 7 millirads/hr. Inside the body of the car, the maximum radiation level is at the driver's feet where it has been found to be about 1 millirad/hr. Film badges have shown that the radiation dose received by the driver of the car has always been less than 50 millirads in any week; this is well below the recommended maximum permissible dose, which is 300 millirads/week[4]. A large drip tray has been fitted underneath the engine to prevent any contamination of the roads by oil leakage, although the activity in the used engine oil is extremely low, usually of the order of 10^{-3} microcuries/ml.

Results and discussion

Two typical wear traces obtained with the car are shown in Figs. 3 and 4. Fig. 3 shows the effect of road speed on wear; the car was driven for about 30 minutes at each speed indicated on the diagram. Speeds were kept as close as possible to their nominal values, but some variations occurred from time to time due to the presence of other traffic on the roads. Increasing the speed from 30 m.p.h. to 40 m.p.h. approximately doubled the wear rate, and at 50 m.p.h. the wear rate was about three times that at 30 m.p.h. Over the final 20 minutes of running, the speed was again reduced to 30 m.p.h. and the wear rate fell once more to its initial level. Fig. 4 demonstrates the manner in which the wear rate varies on starting the car from cold, and the way that this wear is modified by the presence of an anti-wear additive in the lubricating oil.

We are concerned here primarily with the corrosive wear caused by the condensation of acidic combustion products on the cold rings and cylinder walls so, to emphasize the corrosive wear aspect, a modification was made to the coolant system of the car so that the engine took rather longer than normal to warm up. Before the test was started, cold water was passed through the engine's water jacket to ensure that it was quite cold.

The lubricant employed for this experiment was a mineral oil without additives. From the trace it can be seen that, for some time after the starting of the car, a high wear rate prevailed, but as the engine warmed up, the corrosive wear was reduced and the wear rate fell to a lower level. After running for about an hour the car was stopped and the engine again cooled down with cold water. At this stage a small quantity of anti-wear additive was introduced into the lubricant. The cooling was continued for a little over an hour and a second cold start was made. It can be seen that there was not a period of high wear after this start, since the corrosive wear had been suppressed by the additive.

This work has shown that with the aid of radio tracers, it is possible to record ring wear continuously and automatically in moving vehicles. The equipment is robust and reliable in operation and yields easily interpreted records of wear. Thus the technique is particularly suitable for the study of wear under actual running conditions and is able to provide information that is not obtainable at present by any other method.

The authors would like to express their thanks to the Directors of Shell Research Ltd. for permission to publish this paper and to their colleagues at Thornton Research Centre for their help and advice.

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London Shows

COMMERCIAL VEHICLES

A REVIEW of the outstanding features among the exhibits at the Commercial Vehicle Show will be given in the next regular issue of *Automobile Engineer*, published on the 5th November. As in the special issue published annually to cover the private car show, the descriptive matter will be well supplemented by illustrations showing the more interesting details. The price is 3s 6d net, the same as that of the other monthly issues.

PRIVATE CARS

The Show Review number of *Automobile Engineer* will be published on Wednesday, the 26th November. It will constitute a critical review of the more interesting exhibits and will have numerous illustrations of special features and design characteristics. This special issue can be obtained by order from newsagents throughout the United Kingdom, price 3s 6d net. Readers are reminded that it is necessary to make arrangements with a newsagent to ensure that a copy is secured.

Octane Numbers

AN article entitled 'Octane Numbers' was published in the September 1958 issue of *Automobile Engineer*. Towards the end of this article it was stated that, 'Generally speaking side-valve engines with cool manifolds, which nowadays comprise the large majority, give ratings close to Research octane number'. Unfortunately, owing to an omission of a line, this sentence is, of course, incorrect. It should have been, 'Generally speaking, low compression side-valve engines tend to rate nearer to Motor octane number, while efficient overhead-valve engines, with cool manifolds, which nowadays comprise the large majority, give ratings close to Research octane number'.

Malleable Iron

THE booklet 'Modern Malleable', published by Hale and Hale (Tipton) Ltd., of Dudley Port, Tipton, Staffordshire, is now out of print, but a new one entitled 'Malleable Iron' has been introduced instead. It is available for apprentices, engineers, buyers and others interested in blackheart castings. This booklet is well illustrated and comprises sixteen pages of interesting information, including some historical notes as well as technical information concerning the properties, production, heat treatment, fettling, inspection, and control and testing of castings. Patterns, cores, moulds, raw materials, melting and casting are also discussed. The work concludes with some notes on applications.

SKID PREVENTION

Experiments with a Device to Prevent Wheels Locking During Braking

R. D. LISTER, B.Sc., A.M.I.C.E., A.M.I.Mech.E.,* and R. N. KEMP, A.F.R.Ae.S.*

GOOD braking performance is, of course, accepted as one of the essential requirements for safety on the road. The adhesion between tyres and the road surface, however, is not constant but depends on the nature of the road surface, on whether it is dry or wet, and on many other factors. It is also well known that locking of the road wheels by excessive braking is often associated with loss of directional control. For example, in recent on-the-spot investigations of 126 accidents, occurring near the Road Research Laboratory, in which a private car was the vehicle that got into difficulties, there were 13 definite and at least five probable cases of loss of control during braking — a total of 14 per cent. Thus any method of enabling the brakes of a vehicle to be applied up to the maximum adhesion between tyre and road available at the time, without resulting in wheel locking, would be beneficial.

Adhesion between tyre and road

One of the usual methods of expressing the frictional properties of a road surface is to refer to its braking-force coefficient, which is defined as the ratio of the frictional force between tyre and road to the load carried by the tyre. It has been shown[1-4] that the braking-force coefficient is greatest at a particular value of slip between tyre and road: this value is about 15 per cent, but the precise figure is not well established. Therefore, if the brakes of a car could produce this amount of slip at each wheel when braking, the car would stop in a shorter distance than if the wheels were locked. In applying the brakes of a car so as to lock the wheels, the following sequence of events takes place. At the instant when pressure is applied to the brake pedal, the wheels are rotating freely and the slip is zero and, of course, the braking force is zero too. As the pressure is increased, the slip increases until at some critical value the braking force reaches a maximum; if the slip increases still further, the braking force decreases until finally the wheels lock, when, of course, the slip becomes 100 per cent.

The deceleration-time records, Fig. 1(a) and (b), taken on a vehicle with only the front wheels braked and locked, cover the whole range from rolling, zero slip, to sliding, 100 per cent slip, and show a fairly pronounced peak, corresponding to the maximum value of the braking-force coefficient just before the wheels lock. This peak occurs on dry as well as on wet surfaces, but is usually more pronounced on wet surfaces, since they show a much greater difference between the peak and the sliding coefficients of friction. This peak is not always easy to demonstrate when braking on all four wheels since it is difficult in practice to obtain a condition of impending skid simultaneously at all four wheels, each wheel or pair of wheels usually being at a different degree of slip and thereby masking the overall peak for the complete vehicle. Deceleration-time records of braking, in which the wheels did not lock, fail to show this peak, but then the braking force may have been below that for locked wheels: an example of this is shown in Fig. 1(c). Thus, to obtain maximum braking, some means of making use of this peak braking force coefficient must be sought. It has been shown by Bird and Miller[1] that the condition existing when the peak braking force is obtained is unstable and may occur at different values of slip on different surfaces, so it is not practicable to try to maintain the slip at the appropriate value to give peak braking force.

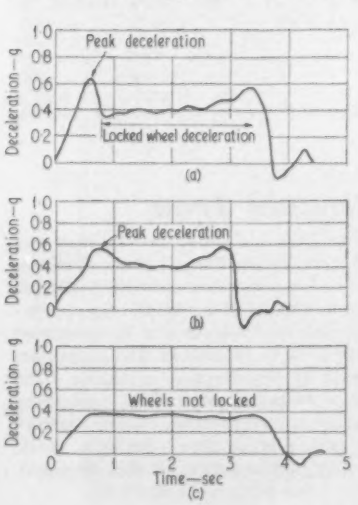


Fig. 1. Deceleration-time curves obtained during braking with the front wheels only. (a) wet concrete surface, initial speed 30 m.p.h.: (b) and (c) dry surface, initial speed 25 m.p.h.

However, by repeatedly applying the brakes and releasing them as soon as the slip exceeds the critical value, it would be possible to pass through this peak braking-force coefficient in a series of repeated applications until the vehicle comes to rest. The device described in this paper works on this principle and in doing so prevents the wheels from locking. It was developed originally for use on the landing wheels of aircraft, mainly to allow the pilot to apply maximum braking on landing without fear of wheel locking and subsequent tyre burst, rather than to take advantage of the peak braking-force coefficients. In preventing the wheels from locking, it might be expected that steering control would be retained and that the loss of control, described by Starks and Lister[5], which is experienced when certain wheels lock, would be avoided.

Description of the Maxaret unit

The Maxaret unit is a device to prevent wheel locking. As employed for early tests, it consists of a rubber-tyred wheel mounted in contact with, and driven by, the rim of the road wheel that it controls; a photograph of a unit is reproduced in Fig. 2(a) and a diagrammatic arrangement is given in Fig. 2(b). The tyred wheel shell houses a drum, which it drives through a clutch mechanism. Surrounding the drum and driven by it, but having a limited independent angular movement of 60 deg is a flywheel rim.

In the steady state, the drum and flywheel are held in driving contact by a control spring. When the angular speed of the road wheel is reduced, the shell and drum also decelerate. The flywheel, however, rotates relative to the drum through the permitted 60 deg of free movement, provided that the deceleration has been high enough for the force exerted by the flywheel to overcome the tension of the control spring. This relative movement between the drum and flywheel operates a valve mechanism to close the pressure supply-line to the brake-actuating cylinder, and at the same time opens an exhaust valve to relieve the pressure already

*Road Research Laboratory, Dept. of Scientific and Industrial Research.

in the actuating cylinder so that the brakes are released and the wheel regains its appropriate angular velocity. Thus, the shell and drum accelerate again, until they overtake the flywheel and reverse the operation of the valve mechanism so that full pressure is again applied to the brakes. This sequence is repeated until the wheel comes to rest. The setting of the control spring determines the angular deceleration that is required before the flywheel will overtake the shell, against the action of the control spring, and operate the exhaust valve.

For operation on the car, the units were set by the manufacturer, the Dunlop Rubber Co. Ltd., to operate when the deceleration of the road wheel exceeded 0·65g. When these units come into operation, they release the hydraulic pressure in the wheel cylinder, so if they should fail to operate, the brakes would still function. Therefore, they can be said to fail safe. A full description of the construction and method of operation of the Maxaret aircraft unit has been given by the Aviation Division of the Dunlop Rubber Company Ltd., in their publication D.M. 1035. A similar unit now being developed specially for use on road vehicles has also been described[6].

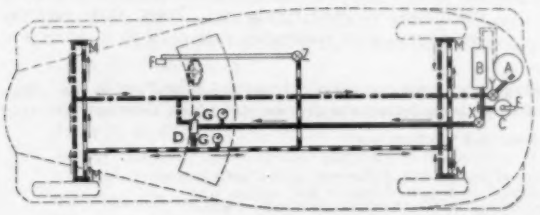
PART I

Tests with aircraft type anti-locking units on a car

The car used for these tests was a 1950, 21 h.p. model of a well known British make, having hydraulically operated two-leading-shoe front brakes and single-leading-shoe rear brakes. Some tests were carried out with the ribbed diamond-tread tyres fitted to all wheels; the remainder of the tests were made with tyres having a smooth tread.

To operate the Maxaret units, the hydraulic braking system of the car had to be modified to permit flow of the brake fluid to take place and to provide a pressurized system with a capacity larger that that of the normal master cylinder.

A diagrammatic layout of the system adopted is shown in Fig. 3, and Fig. 4 shows the actual fittings in the car. Brake



——— pressure supply line; — — — — brake pressure line;
— · — · — · — · brake exhaust line; ▬▬ normal foot brake line

A brake fluid reservoir; B hydraulic pump; C pressure accumulator; D brake control valve; E air charging valve; F foot brake master cylinder; G pressure gauges; H Maxaret units; X, Y and Z shut-down cocks

Fig. 3. Diagram of hydraulic system used in conjunction with the Maxaret units. When the cocks X and Y are open and Z closed, the Maxaret system is in operation: when the cocks X and Y are closed and Z is open, the normal foot brake system of the vehicle is used

fluid is pumped from the reservoir A by an electric pump B to the lower part of an accumulator C. This accumulator is divided in two by a floating piston, the upper part being charged with compressed air from an outside supply to give the required pressure in the fluid. The fluid passes from the accumulator to the control lever D and then to the wheel units M. From the exhaust side of the wheel units and control valve, the fluid is returned to the reservoir. With this arrangement it is, of course, only possible to obtain fully-on or fully-off braking.

By mounting the wheel units so that they could be slid back from the road-wheel rims, thus making them inoperative, it was possible to use this system and obtain locking of the wheels in the tests, the brake fluid under pressure passing straight through the Maxaret units to the brake-actuating

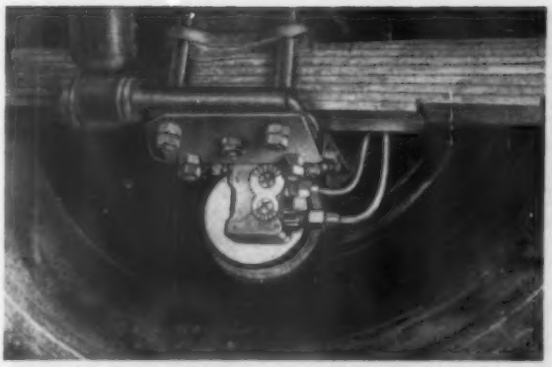
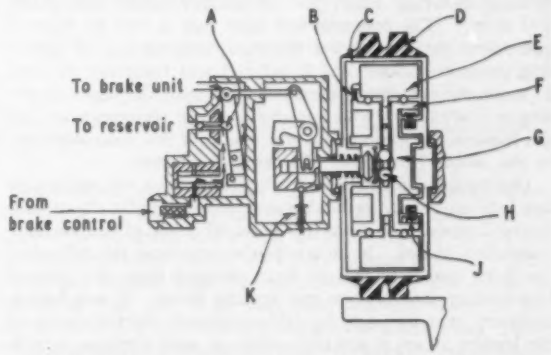


Fig. 2. The Maxaret unit. (a) above, the installation with the eight-lobe cam, round the brake drum, and the follower which can be seen on the crest of the lobe to the left of the Maxaret unit; (b) below, diagrammatic illustration showing the principle of operation of the unit

A exhaust valve, closed; B control spring; C shell; D driving tyre; E flywheel rim; F drum; G cam profile; H thrust balls; J drive spring; K indicator rod



cylinders when the control lever was applied. Shut-down valves were fitted in the system to permit the normal foot-brake system to be used if required.

Braking distances and speeds were measured, using a fifth wheel and counter, Lister and Stevenson[7], while the detailed behaviour of each wheel of the car during the braking period was observed by using a wheel-slip recorder, Lister and Thompson[4]. Fitted to the brake drums were eight-lobe cams, which operated micro-switches; these in turn interrupted light sources in the form of small argon-filled tubes, which were photographed, together with two similar tubes, one controlled by a 1/10 second timing clock and the other by a revolution counter on a fifth wheel towed by the vehicle. The incidence and timing of any wheel locking in relation to distance or time could be observed directly from the photographic record. Typical records are shown in Fig. 5.

Deceleration-time records were obtained during braking, by using a Cambridge recording accelerometer with a modified spring system. A typical deceleration-time record taken with the Maxaret units operating is shown in Fig. 6; it clearly shows the action of the on/off braking.

Description of tests

Preliminary measurements of braking distance, wheel slip, duration of braking, and initial speed were carried out on a wet concrete surface, with the anti-locking device fitted to the front wheels only, the rear wheels of the car being unbraked. These tests were made to find out if the Maxaret units would respond satisfactorily on a road vehicle and to determine a suitable working pressure for the hydraulic system; brake operating pressures ranging from 600 to 1,500 lb/in² were tried. From the results obtained it was clear that there was little practical difference in the response of the units or in the

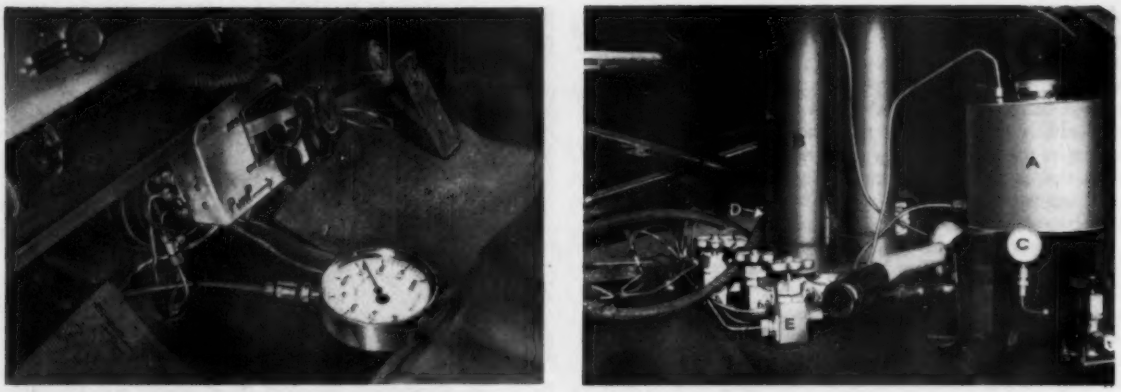


Fig. 4. The two illustrations above show details of the installation of the pressurized hydraulic brake system used to operate the Maxaret units. On the left, the brake control lever and ancillary gauges can be seen, while on the right are the oil reservoir A, the accumulator B, the air pressure gauge C, the hydraulic pump D, and the shut-down cocks E, all of which are mounted in the boot of the car that was used for the skid tests

braking distances when the operating pressure was above 750 lb/in². The response was such that a rate of 6½ to 7 operations per second was obtained, independent of operating pressures above 750 lb/in². It was therefore decided to make the remaining measurements with the units operating at 1,500 lb/in², which is the pressure recommended, by the makers, for aircraft use and which, if the units were not in use, would be capable of locking the wheels.

During these initial tests, the brake-reaction rod anchorage brackets on the car failed because of the rapidly alternating stresses imposed on them by the on/off action of the Maxaret controlled brakes. In an attempt to overcome the difficulty, the front brakes of the car were changed from the original two-leading-shoe type to two-trailing shoes. It was found, however, that without the self-energizing characteristics of the leading shoes, it was impossible on most surfaces, to lock the car wheels even when applying a hydraulic pressure of 2,500 lb/in², which is the maximum working pressure of some components of the Maxaret system. The fitting of Maxaret units with modified valves, giving a less steep pressure rise and fall, reduced the effect of the rapidly alternating stresses. These modified units, together with strengthened reaction rod brackets and the original two-leading-shoe assemblies, were used for subsequent tests.

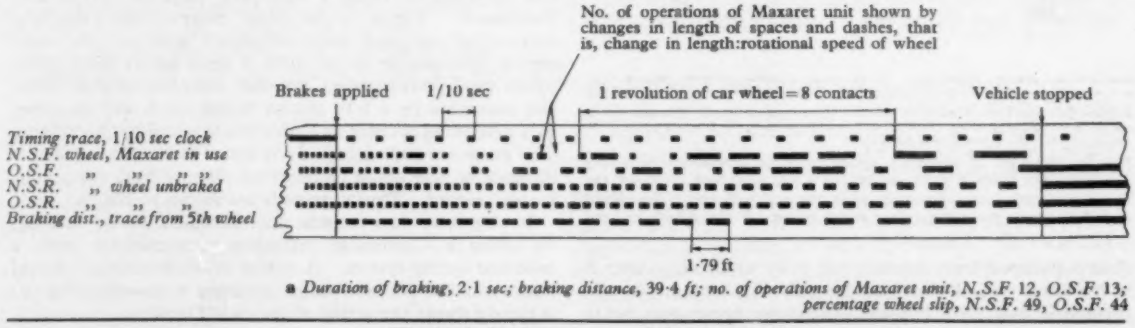
Braking Distances

Nearly all tests were made on wet surfaces, to reduce wear and tear of the car in general and of the tyres in particular. Maxaret units were fitted to all four wheels of the car, and comparative measurements of the minimum braking distances obtainable, both with and without the units in use, were made on five surfaces having different frictional characteristics. The surfaces were as follows:

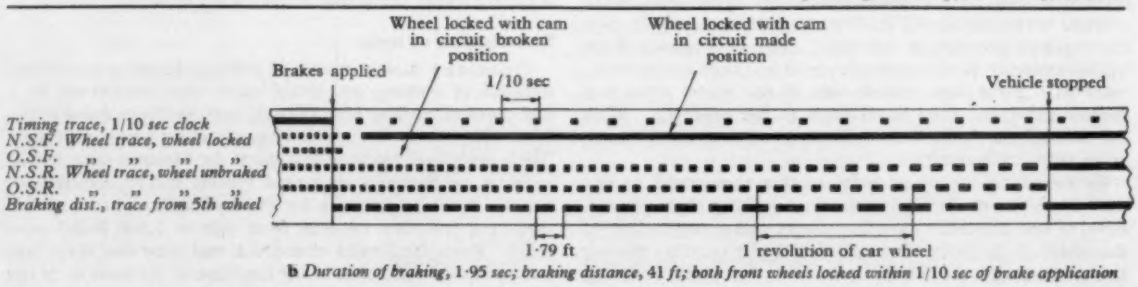
(a) An aerodrome runway, surface-dressed with granite chippings, in good condition. Tests, with pattern-tread tyres, were made over a range of 10 to 60 m.p.h.

Fig. 5. Typical traces obtained from the wheel slip recorder: at the top, with anti-locking device in use on the front wheels only, and the rear wheels unbraked; and at the bottom, with the anti-locking device not in use, the front wheels being locked and the two rear wheels completely unbraked

Key to the abbreviations: **N.S.F.** *near-side front;* **N.S.R.** *near-side rear;* **O.S.F.** *off-side front;* **O.S.R.** *off-side rear*


No. of operations of Maxaret unit shown by changes in length of spaces and dashes, that is, change in length:rotational speed of wheel
Brakes applied
1/10 sec
1 revolution of car wheel = 8 contacts
Vehicle stopped
Timing trace, 1/10 sec clock
N.S.F. wheel, Maxaret in use
O.S.F. " " " "
N.S.R. " wheel unbraked
O.S.R. " " "
Braking dist., trace from 5th wheel
1·79 ft

a *Duration of braking, 2·1 sec; braking distance, 39·4 ft; no. of operations of Maxaret unit, N.S.F. 12, O.S.F. 13; percentage wheel slip, N.S.F. 49, O.S.F. 44*


Wheel locked with cam in circuit broken position
Wheel locked with cam in circuit made position
Brakes applied
1/10 sec
Vehicle stopped
Timing trace, 1/10 sec clock
N.S.F. Wheel trace, wheel locked
O.S.F. " " " "
N.S.R. Wheel trace, wheel unbraked
O.S.R. " " " "
Braking dist., trace from 5th wheel
1·79 ft
1 revolution of car wheel

b *Duration of braking, 1·95 sec; braking distance, 41 ft; both front wheels locked within 1/10 sec of brake application*

TABLE I: Braking distances, on different surfaces, of a car with and without anti-locking device in use on all four wheels

Road surfaces (all wet)	With locked wheels			With anti-locking device			Percentage decrease in braking distance with locked wheels
	Speed m.p.h.	Braking* distance ft	Mean overall deceleration per cent g	Speed m.p.h.	Braking distance ft	Mean overall deceleration per cent g	
(a) Granite-chip surface dressing	20	21	64	20	23	58	10
	30	43	69	30	53	57	22
	50	142	59	50	158	53	11
(b) Rounded gravel asphalt carpet	20	24	55	20	33	41	35
	30	66	46	30	77	39	17
	45	184	37	45	191	36	4
(c) Crushed-gravel surface dressing	20	25	54	20	31	43	24
	30	65	46	30	74	41	14
	45	180	38	45	170	40	−6†
(d) Concrete, test track	20	24	55	20	25	54	2
	30	57	53	30	57	53	0
	45	138	49	45	126	54	−9†
(e) Rock asphalt, slippery track	20	42	32	20	50	27	19

Mean of at least four tests. †Distance with anti-locking device in use shorter than that with locked wheels.

(b) A road having a bituminous carpet with a round gravel aggregate. The surface was in good condition but had a slight gradient. Tests were made, with smooth-tread tyres, at speeds of 20 to 45 m.p.h. During these tests, the deviation of the car from a straight path was also measured, Fig. 10.

(c) A road with a crushed-gravel surface dressing. This was showing some wear in the vehicle paths, but these were avoided as much as possible during the tests. Tests, with smooth-tread tyres, were made at speeds from 20 to 45 m.p.h.

(d) A test track with a concrete surface in good condition. Tests, with pattern-tread tyres, were made at speeds from 20 to 45 m.p.h.

(e) A test track having a special slippery surface of rock asphalt dressed with bituminous emulsion. Tests, with smooth-tread tyres, were made at speeds up to 20 m.p.h.

The results are given in Tables I and II and Figs. 7 and 8.

Frictional properties of the road surfaces

Examination of wheel-slip records showed that the operation of each Maxaret unit was such that the braked wheels passed repeatedly through the point of critical slip, at which the braking-force coefficient is a maximum. It was necessary, therefore, in comparing the frictional properties of the various surfaces, to consider these peak values as well as the locked-wheel values. Deceleration-time records were therefore taken for each surface over a range of speeds with only the front wheels braked and locked. Pattern-tread tyres were used, the surface wetted, and the brakes were applied so that a well-defined peak deceleration was reached between 0·5 and 1·0 sec after starting to apply the brakes. From the records, peak and locked-wheel decelerations were measured; the results are given in Fig. 9.

Steering control

Tests were carried out on the aerodrome runway having a surface dressed with granite chippings, to determine the degree of steering control available with the anti-locking device in use under emergency-braking conditions. These tests consisted of attempting, while braking hard from about 60 m.p.h., to steer the car round rubber marker pylons, spaced 30 yd apart, and trying to turn through 90 deg on left and right-hand bends of 100 ft radius, while applying maximum braking from a speed of 40 m.p.h. The layout is shown in Fig. 11. A 16 mm ciné film was taken of the behaviour of the car when executing these manoeuvres. Tests were also made on this surface to see if steering control could be maintained when the anti-locking devices were fitted to one wheel of each axle, the other wheel being locked; measurements were also made of the braking distances, Table III. It is of interest to note that, with the wheels locked on one side and with the anti-locking device in operation on the other, there was a slight steering response in either direction, but the response towards the locked wheel side was greater than that towards the side on which the anti-locking device was fitted.

Brake-fluid consumption

While the Maxaret units are in operation, actual flow of the brake fluid takes place; measurements have been made of the amounts of fluid used. During the tests of the car on the concrete surface, the fluid that would normally have returned directly from the Maxaret units and from the control valve to the reservoir was collected in a measuring cylinder. The fluid returned during the locked-wheel tests was the amount passed by operation of the control valve alone, since the

Fig. 6. Typical deceleration-time record obtained when braking from a speed of 23 m.p.h. on wet concrete with Maxaret units in operation on the front wheels and the two rear wheels unbraked

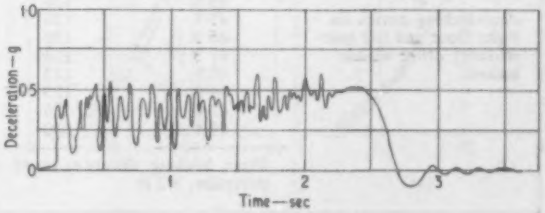


TABLE II: Operation of Maxaret units on different road surfaces—each reading is the mean of at least four tests.

Road surfaces (all wet)	Speed m.p.h.	Mean braking distance ft	Mean overall deceleration per cent g	Number of operations per second of Maxaret unit					Percentage wheel slip*				
				Left front	Right front	Right rear	Left rear	Mean	Left front	Right front	Right rear	Left rear	Mean
(a) Granite-chip surface dressing	20	23	58	5·0	4·0	4·5	5·0	4·5	32	25	26	33	29
	30	53	57	6·0	6·0	6·0	4·0	5·5	35	32	28	33	32
	50	158	53	6·5	5·5	5·5	4·5	5·5	23	27	25	30	26
(b) Rounded gravel asphalt carpet	20	33	41	5·5	5·5	—	5·5	5·5	35	54	32	48	42
	30	77	39	5·0	5·0	5·0	5·0	5·0	31	42	40	44	39
	45	191	36	5·0	5·0	—	5·0	5·0	43	41	26	40	38
(c) Crushed-gravel surface dressing	20	31	43	5·0	5·0	4·5	5·0	5·0	43	40	29	32	36
	30	74	41	5·5	5·5	5·5	5·5	5·5	47	41	32	38	39
	45	170	40	5·5	5·5	5·5	5·5	5·5	43	42	34	32	38
(d) Concrete, test track	20	25	54	7·0	8·0	5·5	6·0	7·0	35	35	26	14	38
	30	57	53	6·5	7·0	5·5	5·5	6·0	38	34	40	16	42
	45	126	54	6·0	6·5	6·0	5·0	6·0	35	40	36	17	42
(e) Rock asphalt, slippery track	20	50	27	4·0	5·0	4·5	4·5	4·5	—	80	78	77	78

*Percentage wheel slip figures are overall values obtained as follows : $\frac{(\text{revs. of rolling wheel} - \text{revs. of braked wheel}) \times 100}{\text{revs. of rolling wheel}}$

Table III : Braking distances and steering control of car with different arrangements of braked wheels

Braking system	Speed m.p.h.	Braking distance ft	Braking distance corrected for 45 m.p.h. ft	Degree of steering control obtained while braking
(i) All wheels locked	45·2	114	113	Nil. Car continued in a straight line irrespective of steering angle of front wheels
	44·1	107	112	
	44·2	107	111	
	45·3	119	117	
	45·2	115	114	
	45·3	114	113	
	Mean braking distance at 45 m.p.h., 113 ft; standard deviation, 2 ft			
(ii) Anti-locking device on all wheels	45·8	133	128	Car steerable between pylons 30 yd apart, Fig. 1, but not 20 yd apart. Car would turn left and right, but breakaway occurred if turn was attempted too sharply
	44·3	127	128	
	45·0	128	128	
	44·9	135	136	
	45·0	139	139	
	44·2	131	136	
	Mean braking distance at 45 m.p.h., 133 ft; standard deviation, 5 ft			Increase relative to locked-wheel distance, 17 per cent
(iii) Anti-locking device on left wheels ; right wheels locked	44·4	121	124	Steering held straight, car went straight. Some response to right-hand turn, very little to left-hand turn
	44·8	131	132	
	44·8	130	131	
	45·0	128	128	
	45·0	127	127	
	45·0	126	126	
	Mean braking distance at 45 m.p.h., 128 ft; standard deviation, 3 ft			Increase relative to locked-wheel distance, 13 per cent
(iv) Anti-locking device on right wheels ; left wheels locked	45·5	129	124	Slight steering response either way but more to left than right
	45·7	131	127	
	45·3	131	129	
	Mean braking distance of 45 m.p.h., 127 ft			Increase relative to locked-wheel distance, 12 per cent
(v) Anti-locking device on right front and left rear wheels ; other wheels locked	45·5	126	124	Initial pull to left, easily corrected. Slight response to either way but more to left than right. If 90 deg turn to left attempted too sharply, car spun through 180 deg
	45·5	126	124	
	46·3	130	124	
	47·3	134	122	
	43·5	123	132	
	44·0	123	128	
	45·0	123	123	
	44·5	120	123	
	Mean braking distance at 45 m.p.h., 125 ft; standard deviation, 3·3 ft			Increase relative to locked-wheel distance, 10 per cent

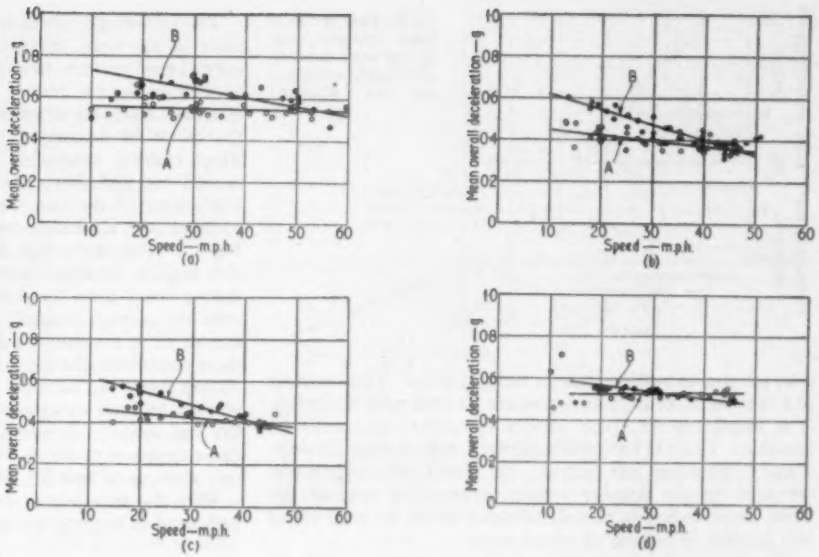


Fig. 8. Four diagrams showing mean overall deceleration from different speeds, A with and B without the anti-locking device in operation, on different road surfaces and with different tyres. These surfaces and tyres are, (a) wet granite chip surface dressing, and pattern-tread tyres: (b) wet rounded gravel asphalt carpet, and smooth-tread tyres: (c) wet crushed gravel surface dressing, and smooth-tread tyres: and (d) wet concrete test track, and pattern-tread tyres fitted to the car

Maxaret units were then inoperative. Additional measurements of the quantities of fluid used by each individual unit and the control valve were made by jacking the car up and operating by hand each Maxaret unit in turn, the exhaust fluid being collected in the measuring cylinder as before. The results are shown in Fig. 12.

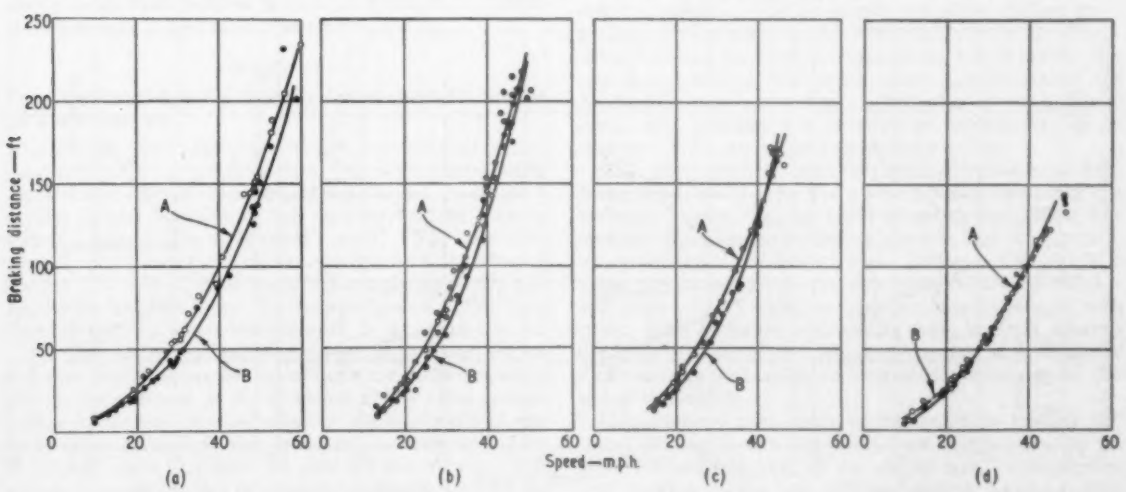
Discussion of results

It can be seen from Table I and Fig. 7 that the minimum braking distances on wet surfaces when the Maxaret anti-locking device was in operation were generally greater than when the car was braked with locked wheels, but that the difference diminished as the initial speed was increased. These differences ranged, for example, from 2 to 35 per cent longer at 20 m.p.h. on surfaces (d) and (b), and from 9 per cent shorter to 1 per cent longer at 45 and 50 m.p.h. respectively, for surfaces (d) and (a). These results are illustrated more clearly in Fig. 8, which shows that on all the surfaces on which tests were made, the mean overall deceleration with locked wheels fell as the speed increased, but with the anti-locking device in use, the reduction in deceleration as the speed increased was almost negligible. It seems, therefore, that above certain speeds, braking on wet surfaces with the anti-locking device might have given shorter braking distances. Figs. 7 and 8 show that shorter braking distances were obtained, by using the Maxaret units, when the initial speeds were greater than about 65, 45, 40 and 30 m.p.h. for surfaces (a), (b), (c) and (d) respectively, the first figure being obtained by extrapolation.

The probable explanation of the superior braking performance with the anti-locking device at higher speeds lies in the fact that the wheels are repeatedly passing through the region of peak braking-force coefficient, Fig. 1, and, as shown in Fig. 9, the ratio of peak deceleration to locked-wheel deceleration is greater at higher speeds because of the fall in the locked-wheel coefficient on wet surfaces, as compared with that on dry surfaces.

Considering in detail the operation of the Maxaret unit it appears that, during the period in which the unit is opening the exhaust valve in the braking system, the brake pressure is

Fig. 7. Braking distances obtained, A with and B without the anti-locking device in operation, during test runs on four different wet road surfaces. From left to right, these surfaces are, granite chip surface dressing, rounded gravel asphalt carpet, crushed gravel surface dressing, concrete track



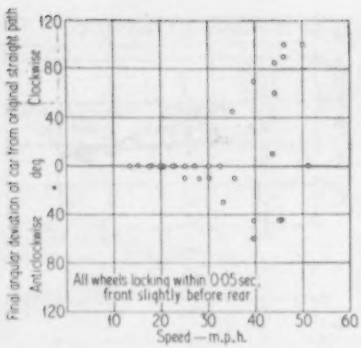

Final angular deviation of car from original straight path
Clockwise
deg
Anticlockwise
All wheels locking within 0·05 sec, front slightly before rear
Speed — m.p.h.

Fig 10. Final deviation from straight path of car when braking with locked wheels on the road surface (b)

released and no braking effort is being applied. Thus, to keep to a minimum the time during which no braking is occurring, it is important for these devices to operate as rapidly as possible. Table II shows that the units were making between 4 and 7 operations per second ; the lowest rate of operation occurred on the slippery surface, presumably because the brake pressure had to be fully released before the road wheel was capable of picking up speed again.

The figures for percentage wheel slip given in Table II refer to the overall values for complete stops. Owing, however, to the method of operation of the Maxaret units, the instantaneous slip may lie between zero and 100 per cent. Further improvement in braking performance could no doubt be obtained if it were possible to design the units to operate over a smaller range of slip, so as to work as long as possible at the value of slip at which maximum braking can be obtained.

Undoubtedly the most striking feature of the tests was the degree of steering control that was retained and the consequent freedom from deviation when braking with the Maxaret units in use at brake line pressures that would normally have caused all wheels to lock immediately. With the anti-locking device on all four wheels, steering was positive, although rather sluggish, and no difficulty was experienced in steering the courses shown in Fig. 11. If, however, too sharp a turn was attempted at high speed, breakaway occurred in just the same way as in normal driving. It was not possible to make any quantitative measurement of the steering control available in these tests ; the only method used was to take ciné films of the vehicle manoeuvres.

The advantages of the steering control available with the units in use were, however, realized better in the tests on normal road surfaces having the usual moderate camber than in tests on flat test tracks. In the locked-wheel tests it was essential, especially when braking from the higher speeds, for the car to be positioned exactly on the crown of the road when braking, otherwise any slight deviation to one side caused it to slide down the camber and to hit the road verge. Deviations of the car, when braking on one of the road surfaces with all wheels locked, are plotted against speed in Fig. 10. This shows that, at speeds above 30 m.p.h., considerable angular deviations occurred. Wheel traces showed that during these tests the front wheels locked simultaneously, with or, perhaps, slightly before the rear wheels ; this is known to be a stable condition[5]. It is probable that under these conditions, the deviation was caused by the front wheels sliding down the camber, the motion of the car then being affected by local variations in braking-force coefficient across the road surface. A turning couple would thus be set up and cause deviation to occur. This deviation did not occur on the two surfaces (a) and (d), which had no camber.

With the anti-locking device in use, no difficulty was experienced in keeping the car straight and in its correct traffic

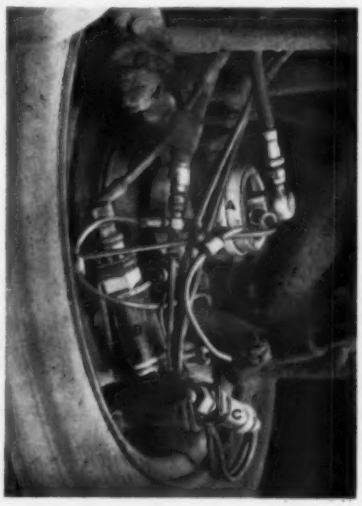


Fig. 13. Installation of the vehicle type, Maxaret unit on the front off-side wheel of a private car

A *Maxaret unit;* **B** *electromagnetic valve;* **C** *pressure transducer*

Fig. 9. Decelerations obtained on different wet road surfaces, (a) to (d), as defined in Fig. 8, with only the front wheel brakes applied

——— *deceleration, locked;* — — — — — *deceleration, peak;* - - - - - - - - - - *peak : locked ratio*

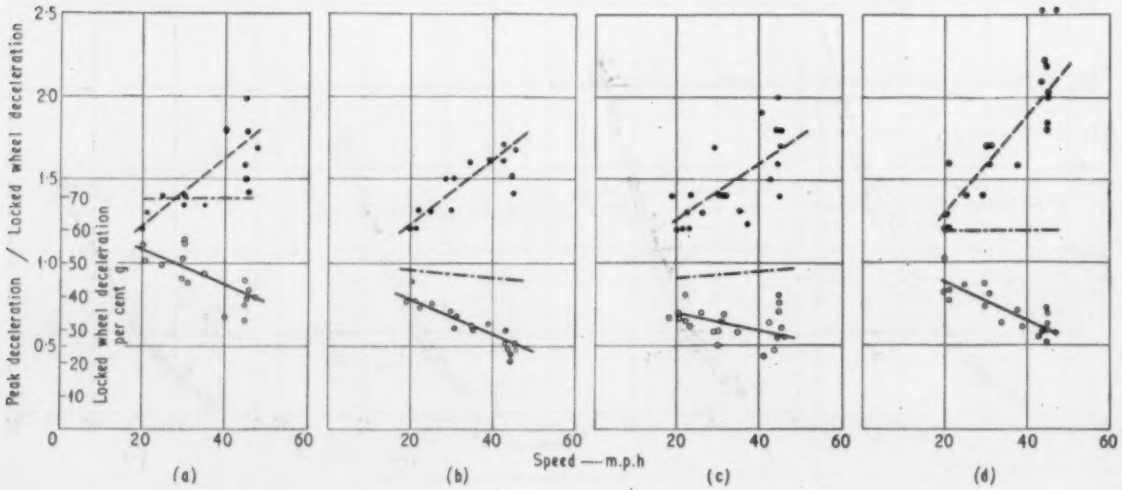


Fig. 11. Right: Diagram showing the layout of the course for assessing steering control while braking with the Maxaret units in use

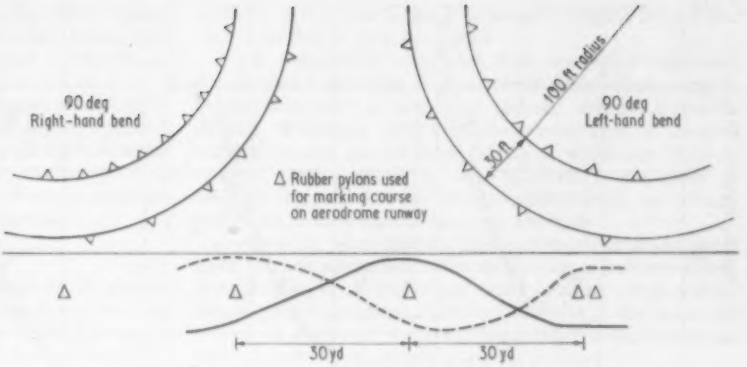
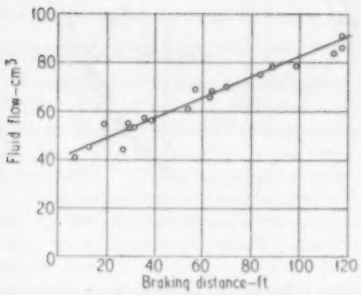


Fig. 12. Relationship between braking distance and fluid flow in the Maxaret system. The initial speeds range from 11-47 m.p.h., and the spill from one operation of the control valve is approximately 20 cm³



lane when braking from any speed. Table II shows that the rate of operation, that is, the number of times the brakes were released per second, varied from about 4 to 8 per second with an overall mean for all surfaces of about 5½ per second. The rate did not appear to be affected by the speed from which the vehicle stopped. On the surfaces that gave a higher mean overall deceleration, surfaces (a) and (d), the rate of operation on individual wheels tended to be slightly higher and on the slippery surface (e) it was slightly lower than this mean. The mean value of the wheel slip varied from 26 per cent to 42 per cent, except on the slippery surface, where it rose to 78 per cent. There was no obvious relationship between the amount of wheel slip and the speed of operation of the units, although both depend on the road-surface characteristics.

It is evident from Fig. 12 that conventional master cylinders would be inadequate for an anti-locking device of this nature. Although each unit required only between 1 and 2 cm³ of fluid per operation of each brake exhaust valve, the total amount required was proportional to the braking distance, Fig. 12. For a braking distance of 100 ft on wet concrete, with Maxaret units operating on each wheel, 80 cm³ of fluid were required, whereas the maximum displacement of a normal ⅞ in diameter master cylinder is only about 70 cm³. Even if a cylinder of sufficient capacity were used, the brake-pedal travel to maintain the required pressure in the system would become excessive. It is therefore necessary with this type of device to use some form of pressurized reservoir, or continuous-flow type of hydraulic power brake system.

PART II

Tests with anti-locking units designed specially for use on road vehicles

In the light of experience gained both from the experiments described in Part I and from their own development work, the Dunlop Rubber Co. have designed and produced a braking system specially for use on road vehicles, incorporating a new design of Maxaret Unit[6]. Tests have been made by the Laboratory, in co-operation with the Dunlop Rubber Co., with a Jaguar Mark VII car having double-pad disc brakes and fitted with this braking system. The front wheel installation and instrumentation is shown in Fig. 13.

The main differences between the redesigned components and those used in the earlier experiments are in the method of driving the unit and in the operation of the valve system. Each is driven from the road wheel by means of a geared ring on the inside of the brake disc, and operation of the unit opens an exhaust valve but does not shut off the supply. The exhaust valve spring also provides the restoring force for the

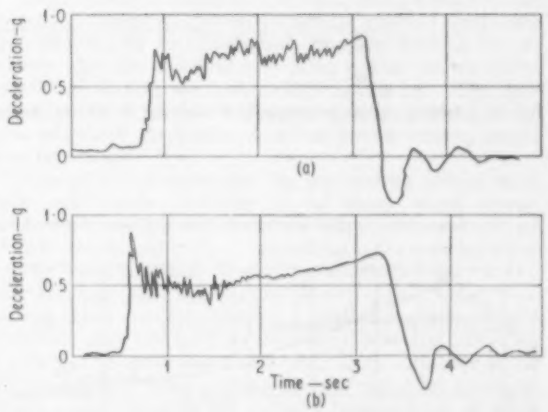


Fig. 14. Typical curves obtained (a) with vehicle type, anti-locking units in operation while braking from 35 m.p.h. on wet concrete, and (b) with locked wheels while braking from 34 m.p.h. on wet concrete

inertia flywheel. A restriction device, known as a modulator, on the pressure inlet side permits an initial flow of brake fluid to take up the brake clearances. This allows full-brake application immediately, and then the flow through the modulator is restricted so that, as the exhaust valve of the Maxaret unit opens, the quantity of brake fluid flowing out is greater than that flowing through the modulator, and the brake-cylinder pressure is therefore reduced. On closure of the exhaust valve, the brake-cylinder pressure builds up again as rapidly as the modulator will allow. These more gradual pressure changes, combined with the absence of self-servo action in disc brakes, obviated the severe judder that was experienced in the earlier tests. Comparison of the deceleration-time curve, Fig. 14, with that shown in Fig. 6 shows that, although the variation in deceleration can be observed, there was not any appreciable judder.

With this system, the brakes are applied by the normal foot-brake pedal, which operates a power valve controlling the hydraulic pressure at the brake cylinders and giving line pressure dependent on the force applied to, and with virtually no movement of, the brake pedal. Although operated by a single control and from a common pressure supply, the rear and front wheel braking systems are independent of each other. Each of the two anti-locking units, however, controls the brake pressure to both wheels of the pair, so operation of a unit reduces the braking on both wheels of the axle at that end of the vehicle.

Measurements were made of the minimum braking distances obtainable with maximum brake application, on the same five surfaces used in the earlier tests. Comparative tests were carried out both with and without the anti-locking

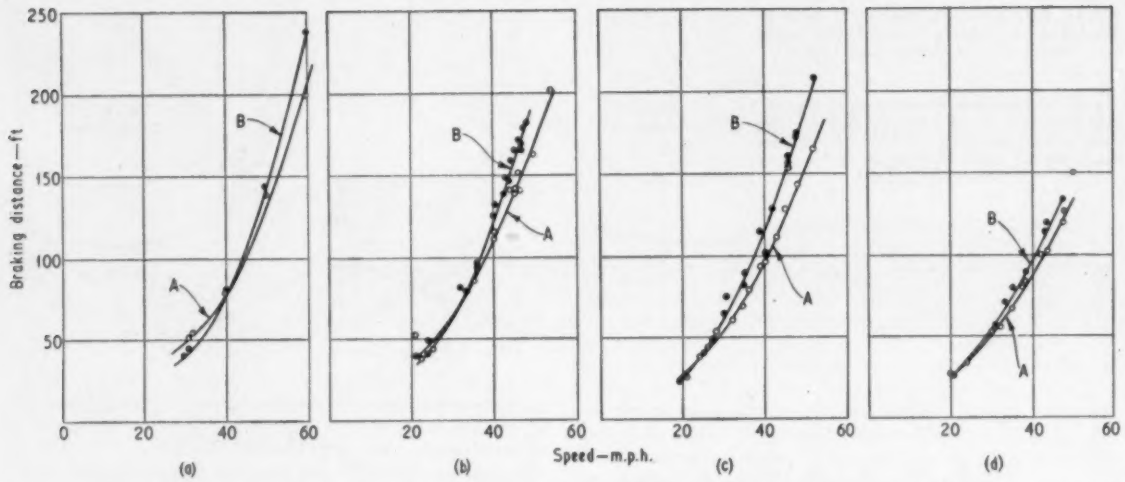


Fig. 15. Braking distances obtained, A with and B without the vehicle type units, on the four different wet road surfaces (a) to (d)

Fig. 16. Below: Mean overall deceleration from different speeds, A with and B without vehicle type units, on the wet road surfaces (a) to (d)

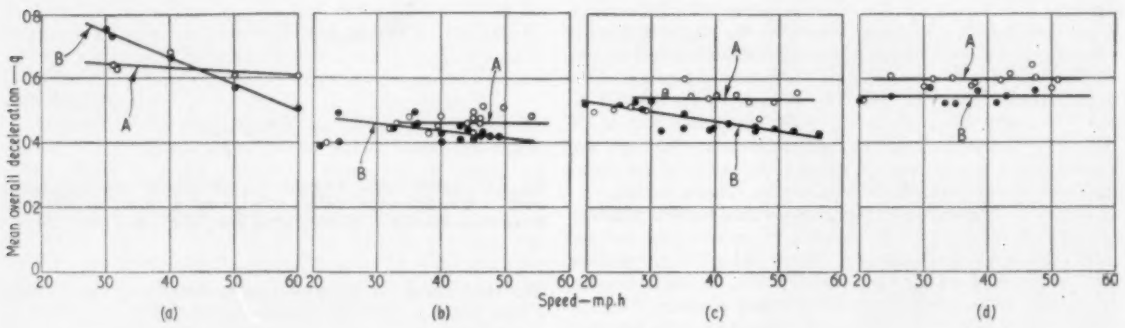


TABLE IV: Braking distances, on different road surfaces, of a car with and without vehicle-type anti-locking units in use—car fitted with disc brakes and pattern-tread tyres

Road surfaces (all wet)	With locked wheels			With anti-locking units in use			Percentage decrease in braking distance with locked wheels
	Speed m.p.h.	Braking distance ft	Mean overall deceleration per cent g	Speed m.p.h.	Braking distance ft	Mean overall deceleration per cent g	
(a) Granite chip surface dressing	30	41*	74	30	47*	64	14
	50	147*	57	50	139*	61	−5†
(b) Rounded gravel asphalt carpet	30	70*	43	30	66*	46	−5
	45	159	43	45	146	46	−8
(c) Crushed-gravel surface dressing	30	53	57	30	47	64	−11
	45	134	49	45	109	62	−18
(d) Concrete, test track	30	47	64	30	47	64	0
	45	127	53	45	121	56	−4
(e) Rock asphalt, slippery track	20	51	26	20	44	30	−13

*All measurements were mean of four tests except those marked **
†A minus sign denotes that the braking distance with anti-locking units was less than with locked wheels.

device in use on a car fitted with pattern-tread tyres. Speed and distance were measured, as before, by a fifth wheel, and deceleration-time curves were obtained with a Cambridge decelerometer.

Table IV gives the minimum braking distances obtained in repeated tests made from speeds of 30 and 45 m.p.h., with and without the anti-locking units in use, for each surface when wet. The braking distances for various speeds are given in Fig. 15, while Fig. 16 shows the overall deceleration obtained with and without the units in use on each surface.

Braking performance

A comparison between the deceleration-time curve, shown in Fig. 14(a), of braking with the vehicle type anti-locking units in use, and that for braking with the original aircraft-type units, Fig. 6, shows the improvements obtained with the newer units. When the original unit operated, the brake pressure periodically fell nearly to zero, with a corresponding fall in the deceleration-time curve, but with the new units, the reduction in pressure, and therefore deceleration, was far less.

In addition, the tests showed a greater improvement in the braking distances relative to the locked-wheel distances. Table IV and Figs. 15 and 16 show that, except for surface (a) at the lower speeds, the braking distance was less with the anti-locking units in use than when braking with locked wheels. The degree of improvement was dependent upon the nature of the road surface and on the initial speed, the greatest improvement being obtained in braking from the higher speeds on the more slippery surfaces.

The results given in Fig. 16 show that on all the surfaces tested, the mean overall deceleration, with locked wheels, fell as the initial speed increased, but remained reasonably constant when the anti-locking units were in use. This feature was also observed in the earlier tests with the aircraft-type units, Fig. 8. Differences between the locked-wheel braking results obtained in the original tests and in these later ones are probably due to several factors, including a further year's wear of the road surfaces, variation in the road surface friction due to seasonal and temperature effects, and to the use of different tyres.

Steering performance

With the anti-locking device on all four wheels, steering control was positive and no loss of steering response, as compared with the unbraked state, was noticed. No difficulty was experienced in steering through the course shown in Fig. 11, provided the speed and degree of turn were not great enough to cause breakaway, as in normal driving. When braking from the higher speeds on road surface (b), which had a slight bend at the end of a limited straight stretch, the car mounted the verge with locked wheels on several occasions, but, with the anti-locking device in use, no difficulty was experienced in keeping it to its proper course on the road.

Concluding remarks

The results described in Part I of this paper clearly indicated the potential value of the aircraft-type Maxaret anti-skid device in providing increased safety when braking in emergencies. Its value arises from the reduced braking distances obtained when braking from the higher speeds, particularly on wet roads, and from the retention of sufficient steering control to enable the driver to keep the vehicle on the road and, in many cases, to avoid obstacles when braking in an emergency. There were, however, certain practical difficulties in applying the device to commercial vehicles and private cars.

In Part II, the results relating to an improved version of the Maxaret device specially designed for use on road vehicles show that the main difficulties can be overcome and that improvements have been made. The result on most road surfaces is that the braking distances obtained are shorter than if all four wheels are locked.

There seems little doubt that, from the safety viewpoint, the Maxaret anti-skid device, particularly when used in conjunction with a disc-type braking system, possesses distinct advantages over the conventional type of coupled braking system, the inherent defects of which are likely to become more apparent as the performance and speed of vehicles increase and as detailed information on vehicle behaviour in emergencies becomes available.

It should be noted that the fitting of anti-locking devices is likely to have advantages other than those mentioned above. For example, in cases of severe brake unbalance arising from lack of proper maintenance or from defects in the brakes on one wheel of an axle, the loss of directional stability that can arise when braking sharply might be avoided.

There would be advantages in fitting anti-locking devices to goods vehicles, even if only on the rear wheels, to overcome the difficulty of providing both adequate braking on all wheels when the vehicle is fully laden and the prevention of rear-wheel locking when the vehicle is unladen. The likelihood of jack-knifing of trailers, owing to the locking of the rear wheels of the tractor or prime mover vehicle, would also be avoided.

Another useful application for anti-locking devices is to solo motorcycles. Locking of the front wheel almost inevitably causes the rider to fall off, and even the very skilled rider has difficulty in controlling a solo machine when its rear wheel is locked. It may be advisable to point out here that although these devices can prevent a road wheel from locking when excessive braking is applied in a longitudinal direction, they can have no effect on a side slip caused by excessive transverse forces that occur when cornering at too high a speed or manoeuvring the road wheel too violently. Many motorcyclists do not use their front brake at all when the road surface is wet, presumably because of the fear of losing control of the machine. The front brake, however, contributes more than the rear brake to stopping in an emergency; consequently any device that will enable the rider to apply maximum braking on the front wheel, confidently knowing that it will not lock, is likely to make a substantial contribution to the rider's safety.

The assistance of the Aviation Division of the Dunlop Rubber Co. Ltd., in supplying the Maxaret units and other components of the system used in these tests, and for advice in operating them is gratefully acknowledged. All the work described in this paper was carried out as part of the programme of the Road Research Board of the Department of Scientific and Industrial Research, and the paper is published by permission of the Director of Road Research.

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OXY-MUFFLER

A Catalytic Exhaust Purifier for Application to Engines Operated in Buildings

EXHAUST fumes may be a serious hazard when petrol engines are operated in enclosed areas. To overcome this problem, Oxy-Catalyst Inc., of Wayne, Pennsylvania, have developed a device, called the Oxy-Muffler, which is claimed to reduce the amounts of organic material and carbon monoxide in exhaust gases.

In the Oxy-Muffler, a catalyst is employed to encourage combination of the exhaust gases with oxygen. This catalyst is a chemical that reacts continuously with the exhaust gases: after the gas reaction is complete, the catalyst reverts to its original state. The reaction of exhaust gases with oxygen, in the presence of catalyst, results in a release of heat, which is indicated on a catalyst activity meter. To start the chemical reaction, there must be some outside source of energy: in the Oxy-Muffler this energy is the heat from the engine exhaust. In fact, the temperature of the exhaust gases must be brought up to 600 deg F. Once started, the catalytic reaction continues in full operation as long as the engine is running.

Three sizes of Oxy-Muffler are manufactured: one for engines with swept volumes of up to 75 in³; another for engines of 75 to 150 in³ swept volume; while the third is for units of 150 to 230 in³ swept volume. All are designed for maximum engine speeds of 3,000 r.p.m.

The engine exhaust gases pass through piping to the venturi air inlet of the muffler. At the venturi, they mix with the appropriate amount of fresh filtered air. Then they enter the catalyst chamber, where the reaction begins; when they leave this chamber, the amount of fumes, odours and carbon monoxide is at a safer level. A stainless steel tube in the catalytic mass forms a well to house a sensing thermocouple to indicate the temperature of the catalytic mass. This indication shows the operator exactly how the catalyst is functioning.

In most applications, the Oxy-Muffler is comparable in size to the existing muffler, or silencer, of the engine. It must, however, be mounted in a horizontal position. The connections are arranged to offer a minimum of restriction to the gas flow, and all the piping is of the same internal diameter as that of the engine exhaust, outlet or down-pipe.

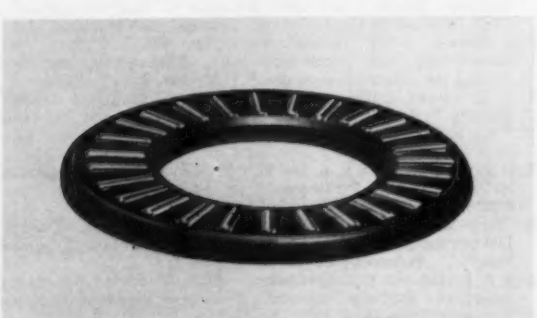
Needle Thrust Bearing

A Nadella Component, Which is Designed for Economy in Respect of Both Space and Cost

AS a result of extensive development work and practical tests, the Société Anonyme des Roulements Aiguilles Nadella have produced a special bearing to deal with axial loads. In this patented design, the assembly comprises a hardened steel retainer with radial pockets for the needle rollers, and a race of hardened steel. The retainer is accurately centred by a lip that is formed round the periphery of the race.

This retainer holds the needle rollers securely in position so that they cannot fall out while the component is handled during the assembly of the machinery in which it is being applied. It also affords positive guidance for the needle rollers, to ensure trouble-free operation. If the needles are not adequately guided, over-heating and wear may occur, because of the setting up of stresses parallel to the axis of rotation; these stresses might be transmitted to the retainer by the needle rollers in such a way that the friction would be increased. To enable the retainer to be of minimum thickness, it is of a slightly dished form, which gives it adequate rigidity. By limiting the thickness of the retainer, space is made available for lubricant.

The Nadella needle thrust bearing has a substantial load capacity, and its requirement with regard to axial space is only 2·8 mm, or 0·11 in



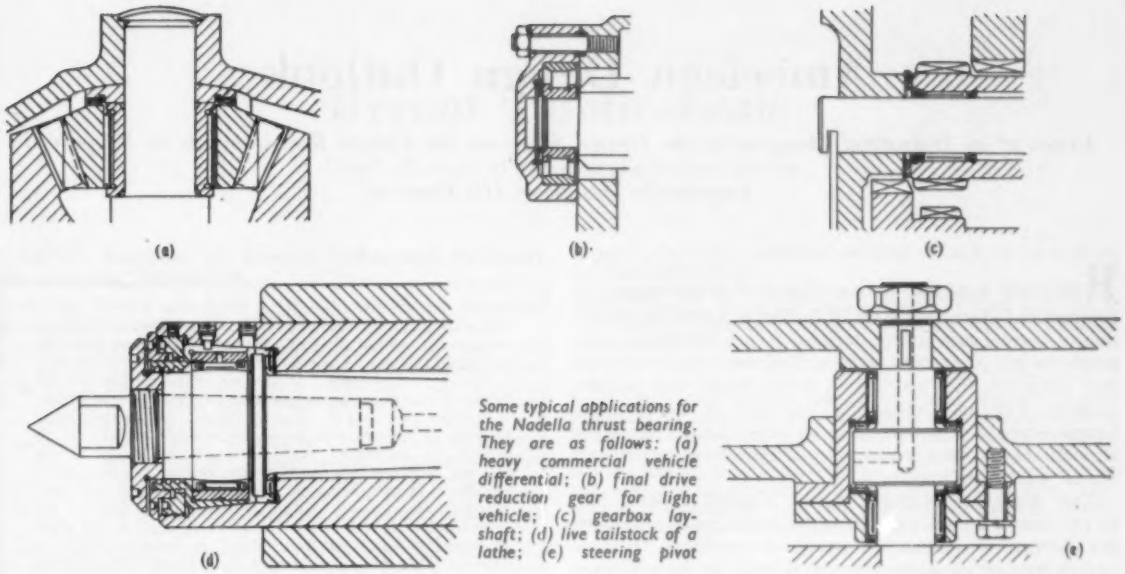
Principal advantages

The principal advantages of the Nadella needle thrust bearing are as follows. For the standard sizes shown in the Table, its axial space requirement is only 2·8 mm, that is, 0·11 in. The bearing has substantial capacity to deal with thrust loads. Its friction coefficient is low, so it can be used for high-speed applications without any danger of its overheating.

The bearing is of robust construction so the lubrication arrangements can be simple. Its cost is relatively low and it is claimed to have a long service life. These bearings can accommodate a limited radial displacement of their shafts, so they can be used in conjunction with bearings designed to take radial loads.

Mounting and assembly

Any plane surface of adequate hardness can be employed as the opposing race for the needle rollers of the thrust bearing assembly; however, this surface must be smooth and machined accurately normal to the axis of the shaft. Shoulders on shafts and pinion faces are commonly employed for



Some typical applications for the Nadella thrust bearing. They are as follows: (a) heavy commercial vehicle differential; (b) final drive reduction gear for light vehicle; (c) gearbox layshaft; (d) live tailstock of a lathe; (e) steering pivot

the opposing races. If the formation of a race on the component is not practicable, a hardened and ground steel disc can be used. This disc can be through-hardened, case-hardened or surface hardened.

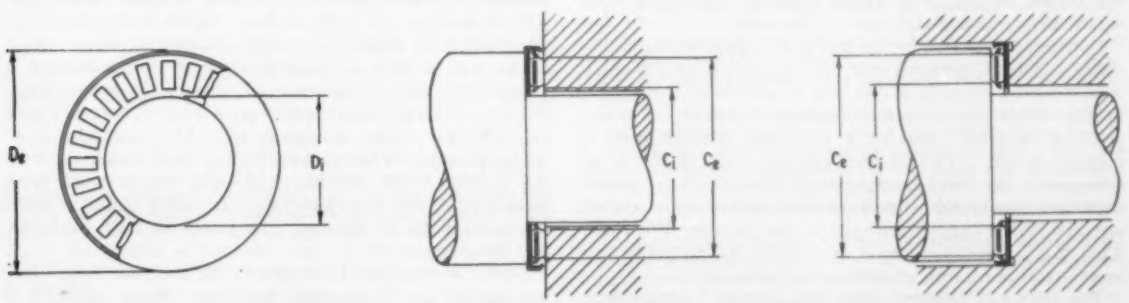
The bearings should be installed concentric to the axis of rotation, preferably being located at their outer peripheries. In some instances, however, it is permissible to locate them at their inner peripheries. As has already been mentioned, the bearing is self-centring so far as relative location of the crown of needles and the race is concerned. This greatly simplifies assembly.

Since the race is only thin, it should be supported by a rigid and continuous shoulder or other abutment face. If the bearing is mounted on a shaft, the diameter of this shoulder should be at least equal to the outside diameter of the race. Where it is mounted in an axial housing or on an external member, the bore of the shoulder should have a diameter that is not less than the internal diameter of the race.

For most applications, lubrication should be effected by a good mineral oil, especially when the running speeds are high. For lower speeds, a medium viscosity bearing grease can be used safely in normal service. It is possible to provide for permanent lubrication that will last the life of the bearing.

Nadella also manufacture special needle thrust bearings for high precision applications. These bearings are of similar design, but have a slightly thicker race, machined to close tolerances. They are used largely to cope with heavy loads or where accurate axial location is required, as in the case of machine tool spindles. Further details can be obtained from Insley (London) Ltd., of 21, Poland Street, London, W.1.

Diagram showing the principal dimensions of the thrust bearing and, below, a table of standard and non-standard sizes scheduled for production



Code	D_i mm	D_i in	D_e mm	D_e in	C_i mm	C_i in	C_e mm	C_e in	Weight lb
				Standard sizes					
AX 1528	15	0·5906	28	1·1024	17	0·6693	24·6	0·9685	0·015
AX 1730	17	0·6693	30	1·1811	19	0·7480	26·6	1·0472	0·017
AX 2035	20	0·7874	35	1·3780	22	0·8661	31·6	1·2441	0·022
AX 2542	25	0·9843	42	1·6535	27·7	1·0906	37·4	1·4724	0·030
AX 3047	30	1·1811	47	1·8504	32·7	1·2874	42·4	1·6693	0·035
AX 3553	35	1·3780	53	2·0866	37·2	1·4646	49	1·9291	0·043
AX 4060	40	1·5748	60	2·3622	43	1·6929	54·8	2·1575	0·053
				Non-standard size					
AX 2241	22	0·8661	41	1·6142	24·8	0·9764	36	1·4173	0·027

American Design Outlook

Views of an Industrial Designer in the United States on the Future Requirements in Respect of Automobile Design in His Country

RECENT reversals in the fortunes of the American automotive industry have set in motion a drastic re-evaluation of current automotive design. Henry Dreyfuss, in an article in the July, 1958, issue of *Consumer Reports*, states that American cars are unsafe, inconvenient and uncomfortable, and that Detroit has lost sight of the needs of people who drive cars. He goes on to say that the appalling accident rate is one indication that designers are dangerously out of touch with reality.

The American National Safety Council reports that 80 per cent of automotive accidents are the result of human, not mechanical, failure. It would be interesting to know exactly how many of these human failures can be attributed at least partly to the fact that the driver's efficiency was impaired by physical discomfort due, for example, to poor seat design, illogical arrangement of controls and inadequate visibility. The author appeals for the introduction of new standards of quality in automotive design. These standards would not only be in the interests of the public, but also might really help to sell cars. Many of the human engineering standards suggested have already been applied by manufacturers of aircraft and heavy commercial vehicles, who have paid considerable attention to the design of really comfortable seats and efficient control layouts. In military equipment, also, where all design ultimately is a matter of life and death, these principles have been applied.

Seat cushions should be firm in the middle and soft at the forward edge. Hard forward edges impair circulation in the thighs and tend to numb the muscles that operate the brakes and accelerator; they may even cause blood clots. Soft-centred cushions tend to cause tiredness by diffusing the weight of the body rather than by concentrating it where nature intended.

Confusion is caused by the fact that control arrangements differ widely as between one type of vehicle and another. A noteworthy example of this lack of uniformity is the hand brake. With regard to foot brakes, the advent of power actuated controls calls for a complete re-assessment of control design. The author suggests that since, in an emergency, the hand reacts quicker than the foot, serious consideration should be given to the employment of a small sensitive brake control adjacent to the steering wheel rim. This, if it is desirable, would, of course, only be practicable in cars equipped with automatic transmissions.

The advent of power steering also calls for a reconsideration of the control problem. Currently, steering wheels not only occupy an unnecessarily large amount of space in the car, but are also a menace in the event of a collision. The small light controls, as used in commercial airliners and military tanks, are worthy of serious consideration.

Push-button controls are being employed increasingly, and, therefore, the danger that the driver may push the wrong button is correspondingly greater. For accurate selection, these push-buttons should be close to the driver's hand, like typewriter keys; also, their locations, shapes, sizes and textures should be standardized so that the driver does not have to look at them, but can keep his eyes on the road ahead at all times while he is operating them.

Henry Dreyfuss has been one of America's foremost industrial designers for almost 30 years. Born in 1904 in New York City and educated there, he began his career as a theatrical designer at the age of nineteen. At the age of twenty-five, he left Broadway and the great success he had earned there, to pioneer in the then new field of industrial design. Among the diverse products of industry and government to which he has lent his talents are telephones, ocean liners, turret lathes, service stations, farm tractors, fountain pens, printing presses, missile launchers, TV sets, aeroplane interiors, thermostats, and troop carriers. He is today a Past President and Fellow of the American Society of Industrial Designers, an organization which he helped to found. He has lectured on industrial design at Harvard, Yale, M.I.T., and before many engineering societies. He is an Associate in Industrial Design at the California Institute of Technology, Visiting Professor at U.C.L.A., and serves on the Visiting Committee of Harvard's Graduate School of Design. His book, "Designing for People", was published in 1955 by Simon and Schuster Inc., 630, Fifth Av., New York, 20.

In connection with the problem of providing adequate rearward vision, the author suggests that the employment of two mirrors is far from satisfactory. Bearing in mind the need for keeping an eye on the road ahead, it is obviously confusing and fatiguing to have to glance continually to a mirror above eye level inside the car and then to another outside. It is suggested that this might be avoided by the employment of a single periscope system to give an unimpeded view of the road behind.

Another measure that is suggested for reduction of muscular, visual and mental fatigue of the driver is international standardization of a system of signal lamps. In this connection, the code of lamp signals commonly used by commercial drivers is worthy of careful study. Both colour and location of the lamps are important because a surprisingly high proportion of drivers is colour-blind. Perhaps the most important part of such a system would be the rear lamp arrangement. The importance of standardization of lamp intensity has been emphasized by the current trend towards large and complex tail lamp arrangements on American cars. In heavy traffic at night the multiplicity of blinding rear lamps is both confusing and dangerous.

So far as cleaning is concerned, the exterior design does not present any insuperable problems. More attention is needed to facilitate the cleaning of the inside of the cars. For example, in cars the floors of which are below the level of the sills, a small cleaning hatch would be most useful; it would then be relatively easy to sweep the dirt out through the hatch instead of struggling to lift it up over the sill. Waterproof materials might be used more for trim, to facilitate cleaning.

The author's statement to the effect that American car designers have fallen into the trap of judging their products solely by their own standards is worthy of serious consideration by manufacturers in other countries. He feels that the men of Detroit have dictated taste while convincing themselves that they have been following public taste.

Recent Publications

Brief Reviews of Current Technical Books

CABMA Register of British Industrial Products for Canada, 1958—1959

London: ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1958. 9¼ × 7. 656 pp. Price 15s.

The CABMA Register has been published annually since 1953 for the Canadian Association of British Manufacturers, whose object is to develop an ever-greater market for British goods in Canada and so stimulate the two-way flow of trade between the two countries. It has proved of service to the Canadian buyer and consequently of value in stimulating Canadian purchases from Britain. In this sixth edition, emphasis has again been placed upon capital and industrial goods of British manufacture required by Canadian industry. All the information contained in previous editions, including that concerning the telephone exchanges, has been brought up-to-date.

A Buyers' Guide is included. It lists alphabetically some 3,200 British products available to the Canadian market, with their suppliers given under each heading. The French equivalents of these headings are set out in alphabetical order in a separate glossary. A directory of nearly 4,000 British firms is given, together with details of their distribution arrangements in Canada, and in another section there is further information about some of these firms and the products they supply. Proprietary names and trade marks are given in special sections, which enable the Canadian buyer to identify products and their sources of supply. The six sections of the Register are easily identified by the slotted index and tinted paper, and instructions are given in French as well as in English on the title pages of each section of this useful work.

The CABMA Register is the Canadian buyers' standard work of reference and the degree of completeness and efficiency attained by this edition has once again been made possible by the full co-operation of the following sponsoring organizations: in Canada, the Dollar-Sterling Trade Council and, in the United Kingdom, the Dollar Exports Council, the Federation of British Industries, the National Union of Manufacturers and the Scottish Council (Development and Industry). Among the contents are: introductory messages and details of the CABMA Organization; Buyers' Guide; French Glossary: Manufacturers and Distributors; Canadian Distributors' announcements; Proprietary names; and Trade Marks.

Motor Vehicle Exercises in Calculation

By H. G. Miles, B.Sc., and L. W. F. Elen, M.Sc.

London: CLEAVER-HUME PRESS LTD., 31 Wright's Lane, W.8. 1958. 7½ × 5. 95 pp. Price 5s. 6d.

The exercises in this book provide a course in calculations to cover the Motor Vehicle Technology syllabus of the City and Guilds examinations in Motor Vehicle Mechanics. In the new syllabus, the work to be done in each of the three years is specified. Exercises 1—25, together with the revision exercises, cover the work of the first year, while exercises 26—38 cover the second year. There are no calculations in the third year. A summary of data and formulae, as well as answers to the problems are given at the end of the book.

Proceedings of the International Conference on Fatigue of Metals, 1956

London: INSTITUTION OF MECHANICAL ENGINEERS, 1, Birdcage Walk, S.W.1. 1958. 11 × 8½. 961 pp. Price 90s.

The International Conference on Fatigue of Metals was sponsored by the Institution of Mechanical Engineers, in co-operation with the American Society of Mechanical Engineers and with the participation of a number of British societies and institutions. It was held in London in September 1956 and in New York in November of the same year. Eighty-one papers were presented in London under five main headings, as follows: stress distribution; temperature, frequency and environment; metallurgical aspects of fatigue; basic aspects of fatigue; and engineering and industrial significance of fatigue. Under the last heading there were three sub-headings, which were: general service, automobiles, and specific components; airframes and engines; and marine engines, railways and welding.

The conference in New York formed part of the A.S.M.E. Annual Meeting, which was attended by nearly 7,000 members of that Society. There were consequently as many as twelve groups of papers being discussed simultaneously. The papers were grouped into six sessions, the subjects of which were as follows: basic studies; stress concentration, combined stress tests, effect of mean stress; cumulative damage, statistical aspect, repeated strain cycling, effect of frequency; effects of temperature, metallurgical aspects, light alloys, corrosion; engineering and industrial significance of fatigue; general service, automobiles, aircraft engines, marine machinery, railways, welding; and engineering and industrial significance of fatigue: aircraft structures. This work is of inestimable value as a collection of scientific treatise of the highest order on all aspects of fatigue.

How to Get Ahead in the Automotive Business

By W. H. Crouse.

London: MCGRAW HILL PUBLISHING CO. LTD., 95, Farringdon Street, E.C.4. 1958. 8 × 5¼. 44 pp. Price 7s. 6d.

This book has been prepared as a guide to the study of the five books of the "Automotive Mechanics Series", most of which have been reviewed in *Automobile Engineer*. It includes chapters on how to study and remember what you read; how to get on in the shop; how to work well with others and be popular; advancing in your profession; and the implications of owning a shop or business.

The Motor Vehicle

By K. Newton, M.C., B.Sc., A.C.G.I., A.M.Inst.C.E., M.I.Mech.E., and W. Steeds, O.B.E., B.Sc., A.C.G.I., M.I.Mech.E.

London: ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1958. 8¾ × 5½. 684 pp. Price 45s.

First published in 1929, this book has been revised many times and is now in its sixth edition. It has become established as a standard work, primarily for students of automobile engineering, but it is also most useful as a work of reference for practising engineers. In the latest edition, some 80 pages have been added, and the entire book has been thoroughly revised. Despite the heavy costs involved in a revision of this type, the price has been limited so that it continues to be within the reach of the average student.

Part I deals with fundamentals of mechanics, and includes a chapter explaining the principles of engineering drawing. Part II is devoted to a study of automobile engines of both petrol and diesel types, attention being given also to matters such as fuels and lubrication. In Part III, gearboxes and other transmission components, axles, brakes, steering, chassis details, and suspension are covered.

The new material includes recent developments in carburettors, including multi-barrel types; fuel and oil pumps; opposed-piston and free-piston engines; gas turbines; supercharging and superchargers, which now are dealt with in a separate chapter; centrifugal and fluid-friction clutches; Manumatic two-pedal control system; Porsche synchromesh gearbox; auxiliary gearboxes and overdrives. The section on torque converters and automatic gearboxes has been expanded into a separate chapter, which includes the following new sections: Chevrolet Turboglide transmission; Borg-Warner transmission; Wilson automatic gearboxes; and MAAG control. There are also new sections on the Garrington constant velocity universal joint; Thornton Powr-Lok differential; Kirkstall double-reduction axle; and the Clayton Dewandre master servo unit. Information is also given on many of the latest engine designs for both private and commercial vehicles.

The book is illustrated with over 600 specially prepared drawings and photographs, which provide a valuable supplement to the text. So far as possible, non-technical language is used and mathematics have been largely excluded. However, from this, it should not be concluded that the work is in any way inferior. On the contrary, it is a sound text book of automobile design practice.

Landing Gear Design

By H. G. Conway, M.A., M.I.Mech.E., F.R.Ae.S.

London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1958. 10 × 6¼. 342 pp. Price 56s.

"Landing Gear Design" is one of the text books published under the authority of the Royal Aeronautical Society. It is a work of the highest technical standard by an author who has had fifteen years continuous experience in the design and production of high quality aeroplane landing gear. Particular attention has been given to advice on practical design features.

Although the book is primarily for professional aircraft engineers, some sections of it are undoubtedly also of interest to automobile engineers. Although few of the general principles could be applied directly to the design of motor vehicle suspension systems, a number of design methods might be studied, with a view to their modification and application for the solution of suspension problems.

The chapters are as follows: Landing gear design; General arrangement of the landing gear; Energy absorption; Tyres; Wheels; Brakes; The general layout of the landing gear; The detail design of landing gear; Shock absorbers; Retraction; Landing gear stressing; and Unorthodox landing gear. This last chapter includes subjects such as snow skis, skid landing gear, track or caterpillar landing gear, and hydro-skis. An index of types of aircraft is included, in addition to a subject index.

How to Write Scientific and Technical Papers

By Sam F. Trelease.

London: BAILLIERE, TINDALL AND COX, 7 Henrietta Street, W.C.2. 1958. 7½ × 5. 185 pp. Price 26s.

In science and technology, the discoveries and advances are of little significance until they are communicated to the rest of the scientific world. Scientists and engineers, therefore, are inevitably faced with the problem of putting the results of their work on paper, and it is most important that their writing should be easily understood. Relatively few people who are not professional writers are able to do this effectively. This book, therefore, is welcome as a medium through which all who have occasion to express themselves in writing may gain a good knowledge of the basic technique of doing so.

The author in his preface, emphasizes the importance of revising manuscripts several times, to make them as nearly perfect as possible. This, indeed, is essential, because a writer cannot be judged on what is in his mind and not properly expressed, but only on what he writes. This may appear to be a glimpse of the obvious, but it is surprising how frequently it is overlooked.

No attempt has been made in the work to include rules of grammar and rhetoric, although reminders are given concerning a few of them—these aspects of writing are treated so well in text books on English composition that their inclusion would be superfluous. Nevertheless, those who are about to prepare a manuscript for publication are advised to study one of these books in addition to the work under review.

Although this work is American in origin, about 95 per cent of it is directly applicable to writing for publication in Britain. The exceptions are obvious: for example, in connection with abbreviations, the reader should refer to British Standard 560, rather than those quoted in the book. The author's advice on the use of colourful words and vigorous terms of expression should be taken carefully, having regard for the fact that such devices are not generally favoured, or at least should be used with discrimination, for the more serious technical works in Britain. This book should be invaluable not only to those who write for publication, but also to those who have to prepare reports of a technical nature.

Commercial Vehicle and Tractor Wiring Diagrams

By E. Mulloy and J. Seale.

London: GEORGE NEWNES LTD., Tower House, Southampton Street, W.C.2. 1958. 9× 6½. 296 pp. Price 30s.

This book, which is a companion volume to "Automobile Wiring Diagrams", is a collection of electrical wiring diagrams for commercial vehicles and agricultural tractors. The aim in its preparation has been at furnishing guidance for auto-electricians, garage mechanics and fleet maintenance staffs in tracing faults in, carrying out repairs to, and wiring renewals of electrical systems. Diagrams of all the leading makes of goods and passenger-carrying vehicles and tractors are given, and they include both petrol and diesel models. More than 40 manufacturers' products are represented, and the number of models covered is over 600. In the introductory section, useful advice on tracing faults in wiring systems is given, and methods of wiring renewals are also discussed. This section occupies the first 14 pages; the remainder of the work is occupied by the wiring diagrams.

The History and Development of Light Cars

By C. F. Caunter.

London: H.M. STATIONERY OFFICE, York House, Kingsway, W.C.2. 1957. 9½ × 6. 120 pp. Price 8s. 6d.

Some thousands of individual designs of widely varying sizes, forms, and technical details, have contributed to the evolution and development of the light, simple and economical motor car as it exists today. The book under review is, perhaps, the first to deal comprehensively with the history of this development. It presents the more important phases of evolution of the light car during the first 60 years of its commercial production and general use. Although the story is necessarily curtailed, the aim has been at presenting a broad and connected survey.

Separate chapters are devoted to the main categories of cars so that the specialized forms of development can be followed clearly from their inception, through their technical evolution and commercial production, to their decline and replacement by other more advanced forms. A bibliography of some of the more important books and references relating to the light car is also given. The work is well illustrated with 92 half-tone illustrations and 24 diagrams. It is both interesting and useful as a work of reference.

Automobile Brakes and Brake Testing

By Maurice Platt, M.Eng., M.I.Mech.E., M.S.A.E.

London: SIR ISAAC PITMAN AND SONS LTD., Pitman House, Kingsway, W.C.2. 1958. 7½ × 5. 97 pp. Price 15s.

This is the second edition of the well known work by Maurice Platt. It is particularly welcome, because of the increasing emphasis that is now being placed on brakes and brake testing. In the work, the author has aimed at presenting a consideration of the fundamental principles of brake mechanisms, as well as the technical information required by automobile engineers and those concerned with servicing. The book has been prepared in such a way that it can be readily understood by readers without specialized technical knowledge.

The chapter headings are as follows: General principles; Brake tests on the road; Garage equipment; Fundamentals of the braking system; Constructional features, adjustment and maintenance; Other types of brakes, servo motors, drums and linings. An index is also included at the end.

Automobile Electrical Equipment

By A. P. Young, O.B.E., M.I.E.E., M.I.Mech.E., F.I.W.M., and L. Griffiths, M.I.Mech.E., A.M.I.E.E.

London: ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1958. 8¾ × 5½. 398 pp. Price 30s.

First published in 1933, this book has long been recognized as the standard work on the electrical equipment of cars, motor cycles, heavy vehicles and other associated applications. Its well deserved reputation as the foremost book of its kind is undoubtedly due to the long and intimate experience the authors have in this highly specialized field, to their ability to express their ideas in clear and simple language, and to their policy of revising the book at regular intervals to keep it up-to-date. As an outcome of this policy they have, in preparing this sixth edition, taken the opportunity of drawing attention to the more recent developments relating to *p-n* junction type rectifiers, electrical control in car transmission systems, and also impulse magnetizers. In addition, a new chapter has been introduced dealing with igniters for gas turbines.

"Automobile Electrical Equipment" is not just a manual on current components. Starting with a lucid explanation of basic electrical theory, it goes on to cover the vehicle's equipment as a whole, and then examines each item in detail, first explaining the fundamental principles underlying the operation and design of the piece of equipment concerned, and then following with practical details. Diagrams and photographs are freely used to make every point absolutely clear. Every aspect of generating, starting, lighting and ignition is dealt with, and information is given on a wide range of electrical accessories, the suppression of radio and television interference, battery charging and on workshop testing and adjustment.

The chapters are headed as follows: Fundamental principles; The complete electrical equipment; The dynamo or generator; The starting motor; The battery; Automobile lighting; Electrical ignition systems; The process of ignition; Battery-coil ignition; The high-tension magneto; Magnetos; Magneto installation; Impulse starters and automatic timing control; Radio interference suppression; Magnetizing and timing a magneto; Sparking plugs; and Igniters for gas turbines. An index is also included. This comprehensive exposition from first principles onwards makes the book invaluable to students in the automobile industry, for whom it is primarily designed, and to the experienced automobile electrician who wishes to bring his knowledge up to date.

Aerodynamic Drag

Analysis of the Factors that Affect Passenger Car Design

R. M. Ogorkiewicz, M.Sc. (Eng.)

CONTINUED interest in economical light-weight passenger cars gives added emphasis to the importance of aerodynamics in vehicle design. By their very nature such vehicles demand the utmost economy in operation and, consequently, the lowest possible resistance to motion, including that contributed by air drag. At the same time, the light weight of these vehicles in relation to their body bulk means that the contribution of the aerodynamic drag of the body to the total resistance is substantial, even at the moderate speeds at which they generally operate, and well worth closer attention.

The general trend in the relationship between body size and vehicle weight is illustrated in Fig. 1, where the ratio of frontal area to weight is plotted against laden weight. This semi-logarithmic plot covers typical designs, ranging from light three-wheelers to heavy trucks, and shows the rising ratio of frontal area to weight with decreasing vehicle weight and, by that very fact, the increasing importance of aerodynamic drag in the performance of light cars.

In the particular case of cars, the relative importance of the increasing ratio of frontal area to weight, and of aerodynamic drag with decreasing weight, can best be shown by considering two vehicles of geometrically similar shape, typical of current practice, but of different size, and comparing the contribution made to their total resistance by aerodynamic drag. Such vehicles may be represented by a six-seater saloon with a frontal area of 22 ft² and a laden weight of 3,500 lb, and by a small four-seater with a frontal area of 17 ft² and a laden weight of 1,700 lb. Apart from the similar general configuration, the two cars may be assumed to have substantially the same specific rolling resistance, expressed in lb per lb of total vehicle weight, which implies, of course, the same tyre pressure, rolling radius and road surface.

Results obtained with these two cars are shown in Fig. 2, where aerodynamic drag is shown as a percentage of the total resistance to motion plotted against vehicle speed. In both cases the drag rises rapidly with speed, but much more so in the case of the lighter vehicle, so much so that, even at 44 m.p.h., it accounts for one-half of the total resistance.

The importance of aerodynamic drag in the case of light cars may be further illustrated by considering a typical four-seater with two different body forms. One is a fairly bluff estate car type, attractive from the point of view of space utilization but indifferent aerodynamically, and the other is a saloon type body of conventional layout but with well rounded contours. In Fig. 3, the expenditure of energy required to propel the two different versions, of the same laden weight and powered by similar engines, is shown in terms of the familiar reciprocal of fuel consumption, in miles per gallon, over a range of steady vehicle-speeds.

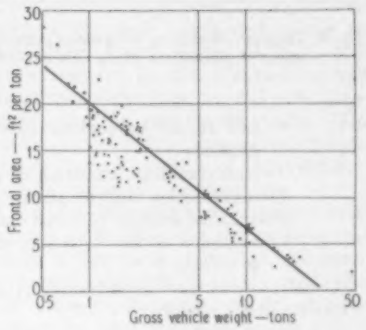


Fig. 1. The general trend in relationship between body size and vehicle weight is shown by the straight line on this semi-logarithmic plot

The differences shown by this comparison are an indication of some of the gains, and losses, that arise from the aerodynamic characteristics of vehicles and which justify close attention in passenger car design. Cars cannot, of course, be designed solely from the point of view of aerodynamics, and the achievement of minimum drag may be overshadowed by other functional requirements. Nevertheless, the aerodynamic aspects of the design and the principal points concerning aerodynamic drag are worth bearing in mind.

General relationships

From dimensional analysis of the variables involved, it can be shown that drag, or force in the direction of relative motion, of a body of given geometrical shape moving relative to air at a particular attitude to the direction of motion is given by the following relationship:

$$D=\frac{\rho l^2 V^2}{g} f\left(\frac{\rho V l}{\mu}\right) \qquad (1)$$

where D=drag, lb

ρ=density of air, lb/ft³, which is 0·0765 at

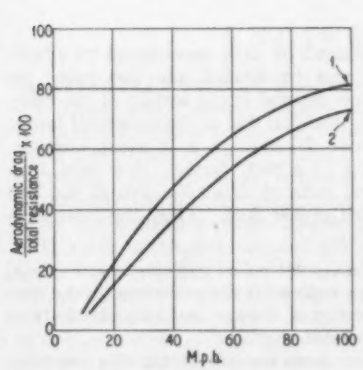


Fig. 2. The aerodynamic drag of two cars, shown as a percentage of the total resistance to motion plotted against the speed of the vehicles

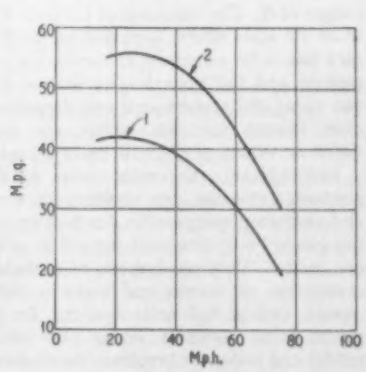


Fig. 3. Variation of fuel consumption with speed of two identical chassis equipped with different bodies. Both were loaded to give the same kerb weight

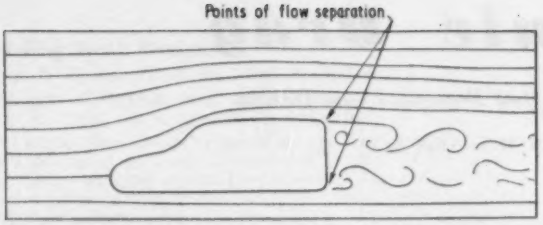


Fig. 4. Diagram illustrating the general pattern of eddies set up behind bodies of most of the estate cars and commercial vehicles

14·7 lb/in² barometric pressure and 60 deg F ambient

l=characteristic dimension of the body, ft

g=acceleration due to gravity, ft/sec²

V=relative velocity between the body and air, ft/sec

μ=viscosity of air, lb/ft-sec, which is $12{\cdot}18\times10^{-6}$ at 60 deg F

Usually, however, aerodynamic drag is expressed in a different form, namely:

$$D=\frac{C_D\rho A V^2}{2g} \qquad (2)$$

where C_D is the drag coefficient and A, which takes the place of the square of the characteristic dimension, is the projected area in the direction of the vehicle longitudinal axis.

Automotive engineers in this country and the United States have used yet another expression for drag:

$$D=K A V^2 \qquad (3)$$

Although ostensibly simpler, equation (3) is only another form of equation (2) and by comparison suffers from the serious disadvantage that whereas C_D is a non-dimensional constant, the factor K is not and depends on the units employed, as well as the air condition, that is, its density. Moreover, this form of expression is out of step with the vast body of aerodynamic practice and, of the two, the presentation of drag in the form of equation (2) is to be preferred. From (2) and (3) it is clear that:

$$K=C_D\frac{\rho}{2g}$$

or, $C_D=391\ K$

at 14·7 lb/in² and 60 deg F, when V is in miles per hour.

Scale effect

Comparison of equations (1) and (2) show that C_D is a function of $\frac{\rho Vl}{\mu}$, the non-dimensional Reynolds Number R, which represents the ratio of inertia forces to viscous forces. In general, C_D decreases with R, although the degree of this variation may not be marked over considerable ranges of R. The variation of C_D with R, commonly referred to as the scale effect, accounts for the fact that considerable care has to be exercised in comparing results obtained with models and full scale bodies: unless R is the same in the two cases the systems are not dynamically similar. Thus, even though the two bodies are geometrically similar, different values of C_D will be obtained.

Unfortunately, the same value of R is not, in itself a guarantee that the two systems are similar in every sense, and therefore comparable, for it does not take into account the general turbulence of the air in which the observations are made. Varying degrees of turbulence complicate the correlation of results and make it difficult to extrapolate model tests to full scale bodies. In practice, however, a much more common reason for discrepancies between model and full-scale results is the degree of detail finish and accuracy of the model, particularly with regard to small protuberances, which are inseparable from an actual vehicle but which are not always reproducible on a model. Another important factor is the method of mounting the model in a wind tunnel: this involves the correct reproduction of the road surface influence on the air flow past the car. As a result of such practical influences, model tests are apt to give values of C_D considerably lower than those realized with full-size vehicles.

On the other hand, over the range of conditions and Reynolds Numbers encountered with actual cars, C_D is virtually constant, and may be treated as such from the design point of view. As a matter of interest, typical values of Reynolds Number in the range of 30 to 90 m.p.h., based on overall length as the characteristic dimension, vary from about 4×10^6 to 17×10^6.

Drag coefficients

Actual values of C_D for a number of different bodies have been collected by Koenig-Fachsenfeld,[1] who also gives much detail of earlier, mainly German, work on aerodynamics of cars, while Reid[2] has listed results for several more recent cars.

A typical value of C_D for the currently popular slab-sided type of car with reasonably rounded contours and stepped rear is 0·45; this figure actually applies to the the two cars illustrated in Fig. 2, while 0·65 and 0·35 correspond to the alternatives in Fig. 3. Several of the aerodynamically superior cars have drag coefficients of only 0·2 to 0·3; the Panhard Dynavia experimental saloon,[3] exhibited in 1948,

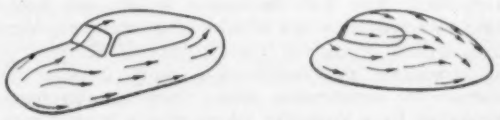


Fig. 5. Diagrams showing the three-dimensional nature of the flow over a vehicle body. It is important that the contours of the body be arranged to give a smooth transition from one surface to another

had a C_D as low as 0·17. Even better results have been achieved in a few special cases: for instance, 0·1 has been claimed for the Tarf, the twin torpedo shaped vehicle built in Italy by Taruffi. For comparison, it may be of interest to add that the fuselages of typical subsonic aircraft have drag coefficients of the order of 0·15 to 0·17 and trucks, on the other hand, between 0·7 and 0·9.

Comparison of drag coefficients is interesting from the point of view of grading of cars with respect to aerodynamic drag and showing the degree of possible improvement. In itself, however, it gives no indication of how such improvement may be achieved. For this it is necessary to consider the problem in greater detail and examine the sources of aerodynamic drag.

Drag Components

In general, the resistance or drag experienced by a body moving through air may be divided into two parts: the resistance due to forces normal to the surface of the body, or pressure resistance, and that due to the tangential forces, or frictional resistance. In practice it is usual, however, to sub-divide the drag of a body further, into form drag, skin friction drag and induced drag, the sum of the first two terms being called profile drag. Thus, the total aerodynamic drag of car is:

$$D=D_{form}+D_{friction}+D_{induced}+D_{internal} \qquad (5)$$

where the fourth term represents the resistance to the flow of air through the interior of the car, resulting mainly from the requirements of engine cooling.

Induced drag results from the generation of a net force

component normal to the direction of motion, or lift, which, in the case of a three-dimensional body, gives rise to an associated force component along the axis of motion. The latter, which is the induced drag, is fortunately relatively small, being of the order of 10 per cent of the total drag for typical cars. To reduce it further requires alteration to the shape of the body, and, consequently, pressure distribution over it, to reduce the lift and, with it, induced drag.

Skin friction arises from viscous stresses in the thin layer of air adjacent to the surface of the body. It is generally dependent on the total surface area of the body and the nature of the boundary layer, which may be laminar or turbulent, the latter resulting in a higher drag than the former; surface roughness favours transition from the laminar to the turbulent condition and also increases skin friction by the finely divided pressure drag of the surface irregularities.

If a laminar boundary layer exists on the forward part of the body, as it might in the absence of strong turbulence in the air, the transition to a turbulent boundary layer is affected by the shape of the body, so to that extent, the division of profile drag into skin friction drag and form drag is not strictly accurate, for it implies that body form has no effect on skin friction. In practice, however, this division is largely correct, and in the case of cars the predominant effect of body shape is on form drag.

Skin friction drag itself is relatively small, of the order of 10 per cent of the total drag for current car types, but its importance increases as the body shape becomes more streamlined and the drag coefficient decreases. For bodies such as aircraft fuselages, for instance, it may account for about 75 per cent of the total.

Flow separation

In the case of most cars, however, both skin friction and induced drag remain of relatively small significance, and the largest and most important component of the total drag is form drag. The latter arises from the separation of flow from the surface of a body at abrupt changes in the contours, or, more precisely, whenever the local velocity near the surface is appreciably reduced and, because of the consequent backward flow of the inner part of the boundary layer, eddies are formed. The eddies, or wake of vorticity, set up behind a bluff body are indicated diagrammatically in Fig. 4, and the energy that they consume accounts for at least one-half of the total drag.

Reductions in the form drag of a motor car body are to be sought primarily at the rear, because the front, provided it is well rounded, has little effect on it. To reduce separation to negligible proportions would require a very slow tapering off and an impossibly long body. Nevertheless, even if the actual point of separation cannot be moved, some improvement can be obtained by more rapid taper or partial extension of the body into the region of the wake and thereby reducing the intensity of the eddies. From this point of view, the once popular sloping back is particularly attractive, but even the stepped rear of the typical present-day saloon gives better results than the abrupt termination of the estate car or van bodies.

A major contribution to form drag is made by protuberances such as headlamp hoods, prominent channels over the windscreen, and by registration plates. As a rough approximation, the resistance of such parts is proportional to the angle through which their edges deflect the air stream; consequently, their shape results in high drag coefficients. Moreover, they are located in regions of high local velocities; this increases the drag, as also may interference with the flow over the rest of the body. The underside of the average car, with all its exposed mechanical components, provides a further example of the protuberances which contribute substantially to form drag.

Consideration of the two-dimensional flow past a body can do much to improve form drag, but the cross section in side elevation, which is frequently given the most attention, is by no means the only profile to be watched. Air flow over the body is, of course, three-dimensional, and this is made particularly clear by the direction of the streamlines from the top of the bonnet on to the sides, as indicated in Fig. 5; this calls for a smooth transition from one surface on to the other to avoid local flow separation.

The fourth component of the total drag of a car, internal flow resistance, arises from the losses in the air flow through the interior of the body, mainly for engine cooling. In general, it is of the same order as, or slightly greater than, skin friction or induced drag and is susceptible to considerable reduction by careful design of the ducting. Because of this, and the fact that alterations to the internal flow system need have little effect on the appearance of the car, it is the most obvious and, except for cost, the easiest field for design improvements.

Rewards of reduced internal flow resistance, in terms of smaller overall drag, however, are likely to be small in relation to those that would accrue from the reduction of form drag possible with the great majority of present-day cars. Form-drag is undoubtedly the principal factor contributing to the high total drag of vehicles; the reduction that can be effected is chiefly due to diminishing the eddies formed in the wake of the car and the prevention of local flow separation, to which many present-day styling features are all too conducive.

References

1. R. Koenig-Fachsenfeld: "Aerodynamik des Kraftfahrzeugs", Umschau Verlag, 1951.
2. J. P. M. Reid: "Body Form", *Automobile Engineer*, Vol. 44, January 1954.
3. M. Victor: "La Nouvelle Dyna", *Revue d'Aluminium*, 1953.

ENGINEERING DRAWING PRACTICE

A SPECIALLY abridged, low-priced version of B.S. 308, Engineering Drawing Practice, has been produced by the British Standards Institution. This version is B.S. 308A, and it is important to indicate the suffix A clearly when ordering copies. The new booklet is intended for students who are in the early stages of an engineering education; its price is 4s 6d, less a discount of one-third for students.

This new publication contains much of the information in the complete edition, and it occupies 40 pages. Each page carries a cross reference to the equivalent page of the earlier edition. Among the subjects it deals with are: scales, types of lines, projection, lettering, sectioning and sectional views, conventional representation of common features and materials, and abbreviations. The following information is given, also, concerning dimensioning and tolerancing: principles, projection and dimension lines and leaders, dimensions, methods of measuring common features, and keyways.

The parts of B.S. 308 excluded from this abridgement are those dealing with drawing office procedure, the layout of industrial drawings, and geometrical tolerancing—all subjects not generally taught until a higher stage of study is reached. However, as is pointed out in the work, it is important to introduce, at the earliest possible stage of the course, the idea of geometrical tolerancing and its problems, and this principle has been followed. Copies can be obtained from the British Standards Institution, 2, Park Street, W.1.

THE DE HAVILLAND CABLE TESTER

An Electronic Instrument for the Rapid Checking of Multi-core Cables or Electrical Systems for Continuity and for Insulation

INITIALLY designed for checking relatively complex electrical systems in aircraft, this instrument is of value for any type of installation where cabling is already fixed in position or where the cables are run in confined spaces. Thus its application extends to all aspects of factory or plant maintenance; control and actuation circuits for machine tools and transfer equipment, furnace and process controls, and all usual plant electrical services. It can also be bench mounted and used in production for checking pre-formed electrical harness or wired assemblies before dispatch or installation.

The manufacturers, de Havilland Propellers Ltd. of Hatfield, Hertfordshire, are now producing the instrument for general industrial use. A range of couplers and ancillary equipment is available to make the instrument adaptable to a wide variety of applications, and special fitments can be designed and produced to meet specific special requirements. Power is supplied by one of two standard packs clipped to the bottom of the instrument and serving as a base. For the fully portable unit the pack houses a 24 V lightweight secondary battery, of the type having no free electrolyte, of capacity sufficient for about sixteen hours of normal operation. For use on the bench or where a mains supply is conveniently available, an A.C. power pack, 110-250 V, 45-65 c/s, single phase, is provided.

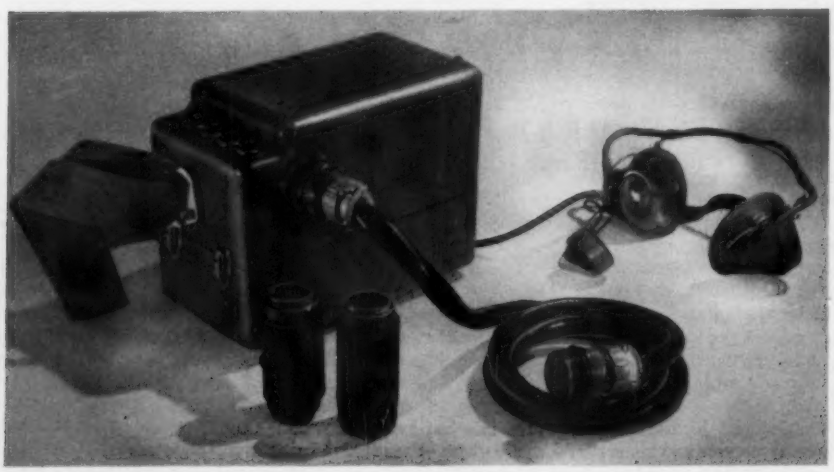
Since the packs are interchangeable it is possible to use a single instrument with a mains pack on the bench for regular service in production and to substitute the battery pack when it is required for maintenance operations out of normal shift hours, for use in remote locations, or for checking or diagnosis of faults on vehicles on the road or equipment in the field.

Either a continuity check or an insulation check can be carried out from one end of a cable run. The remote end is terminated by a small unit weighing less than half a pound. Connector capacity is up to 25 ways per test. Larger cables or bunched conductors can be accommodated by the use of additional adaptors. Continuity checks are made with closed circuits and insulation checks with open circuits. In the case of vehicle wiring, a continuity check would be made with all circuits closed and the test voltage will be sufficiently low to obviate any possibility of damage to light and signal bulbs or other equipment. For insulation tests on a vehicle, the circuits could readily be broken at the fuse board and connections established with a suitable fan-strip type of coupling device. The insulation test voltage is 500 D.C. but this is not uncomfortably shocking and is positively non-lethal owing to the current limitation.

A test is initiated by a press button and the instrument automatically selects each wire in turn. The indentification of the particular core undergoing test is indicated by a letter on a rotating drum, which is displayed in a window in the top surface of the instrument. This is illuminated by a red light and the selector is stopped when a failure is located. The sequence can be continued by pressing a restart button. Successful completion of a test sequence is notified by the extinction of a separate indicator lamp, and a reset button prepares the instrument for a further test sequence.

A single outlet from the instrument can be adapted for use with any cable by means of a suitable coupler to meet requirements. Where a connector has a multi-position keyway, as on the Plessey Mk IV, for example, the coding is checked by a clearly marked movable keyway on both the coupler and the termination unit. For use in locations difficult of access, where the test set cannot be brought closely to the connection, an extension cable can be fitted.

On continuity checks the speed of operation is 5 conductors per second. On insulation tests each conductor is checked to all others bunched and to earth. The time for each conductor can be arranged, as required, at 6 sec, 15 sec or 30 sec. Current consumption will fluctuate according to the state of test but under all conditions is less than 30 watts. Tests can be conducted on cable runs of several miles and the set is capable of testing up to 50 miles. For such long distances the threshold level of the instrument would be raised to suit. This would be effected by a factory re-setting, since it is deemed inadvisable to provide externally



The de Havilland portable cable tester, equipped with its connector cable, termination unit, and intercommunication headset

Weighing only 19 lb the set is easily portable and can be operated while suspended by a sling

A test set is here shown being used to check the wiring of an electronic computer

adjustable controls. In usual operation it is essential that the triggering level is maintained at a constant value.

To facilitate the testing of long cable runs where the termination unit is more conveniently attached by a second operator, an intercommunication amplifier is incorporated in the instrument. Connection is established by way of the cable under test and standard Service-style headsets may be used, as illustrated. When this system is in use both operators receive additionally an audible signal as each individual test is made and can distinguish when a cable failure is detected.

In its portable form the overall dimensions of the set are 11 in × 6 in × 8½ in and the weight is 19 lb. Constructionally, all electronic components are designed to Government specifications and are housed in a stout glass fibre-reinforced plastics housing. It is provided with a webbing strap that can serve either as carrying grip or as a neck sling when operating. A selection of couplers and termination units can be carried in a pouch belt for field use.

A transit case, also of glass fibre-reinforced plastics, provides storage space for a complete set of test equipment and accessories. A charging set can be supplied for maintaining the battery in good condition between periods of service. It is fitted in the transit case and connections are made when the test set is placed in position in the case. Internal regulation ensures that in four hours the battery will be fully charged, but there is no risk of overcharging however long the equipment remains out of service. A charging-current meter provides an indication of the battery condition prior to disconnection of the test set from the charger and removal from the case.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

OCTOBER

Birmingham

Monday 13th October, 6.30 p.m. at the Birmingham Exchange and Engineering Centre. Paper: "An Investigation into the Mechanism of Oil Loss Past Pistons", by Dr. P. de K. Dykes, M.I.Mech.E.

Coventry Graduates

Thursday, 16th October, 7.30 p.m. in the Grosvenor Room, Leofric Hotel, Senior Member's Address, by W. M. Heynes, M.I.Mech.E.

Luton

Wednesday, 8th October, 7.30 p.m. in the Assembly Room, Luton Town Hall. Centre Chairman's Address entitled "Some Experiences during Forty Years of Automobile Engineering".

North-Eastern

Wednesday, 15th October, 7.30 p.m. in the New Lecture Theatre, The University, Leeds. Paper: "Suspension of Internal Combustion Engines in Vehicles", by M. Horovitz, A.M.I.Mech.E.

Scottish

Monday, 20th October, 7.30 p.m. at the Institution of Engineers and Shipbuilders, Glasgow. Automobile Division Chairman's Address entitled "Years of Development".

Western

Thursday, 9th October, 6.45 p.m. at the Royal Hotel, Bristol. Automobile Division Chairman's Address entitled "Years of Development".

B.M.C. Engine Replacement Plant

An Output of 4,000 Units per Week is Planned for the Largest Specialist Plant in the World

UNTIL recently replacement engines and gearboxes for Austin and Nuffield vehicles were produced in replacement plants at Longbridge and Coventry. These activities have now been concentrated in a single huge plant, the largest of its kind in the world, occupying an area of 15,000 square feet in the two largest workshops of what was formerly the Riley works, at Dunbar Avenue, Foleshill, Coventry. The plant is operated as a subsidiary of the main factory of Morris Motors Ltd., Engine Branch, at Courthouse Green, Coventry.

Apart from new reception and dispatch bays and a modern administrative block, there has been little major structural change. Careful planning has made it possible for all the tools and production equipment involved in this gigantic reorganization to be rehoused in existing buildings. The plant includes five assembly lines withdrawn from the Morris factory at Courthouse Green and from the Austin works at Longbridge. Incidentally, this transfer of plant from the Morris factory has freed space which is now equipped and utilized to effect a desired increase in the production rate of current-model engines and gearboxes.

Output at the new replacement plant is planned for 4,000 units per week. This figure is approximately twice that for the combined output of the earlier Longbridge and Coventry plants. Currently the output is running at 2,500 per week and the larger figure will be attained by two-shift operation. Production planning and operation is necessarily most comprehensive, since the total includes units for most of the Austin and the Nuffield vehicles on the roads today. It is still possible to supply replacement engines for some vehicles manufactured more than twenty years ago. The extreme instance is a vehicle built in 1935. In all, 147 different models are catered for, so the complexities of the organization can be appreciated. Although the total number of replacement units produced will increase, no doubt the number of variant models will gradually decrease as the results of the policy of type reduction in current-model production become evident.

The Replacement Unit Scheme provides for the acceptance of worn engines and gearboxes and the supply of replacement units that have been completely rebuilt from reworked and new parts to standards of accuracy, tolerance, assembly, inspection and testing similar to those accorded to new production units for current models.

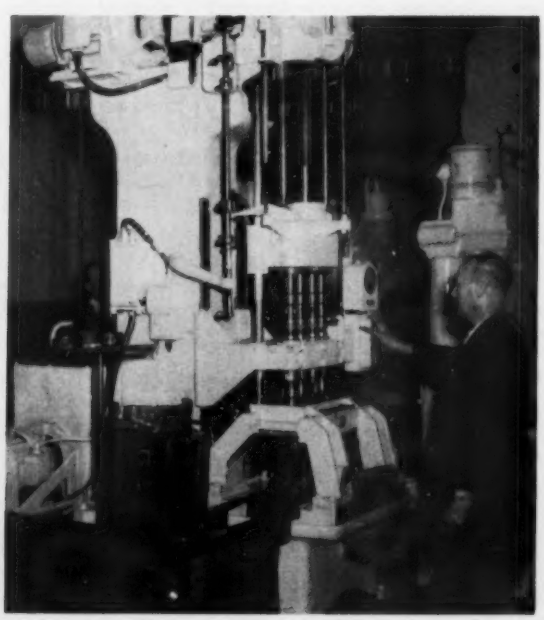
An engine is received at the plant after years of service, and is first examined by an experienced inspector to ascertain that it has not suffered gross damage from abuse or neglect, or from impact in an accident. It is then marshalled with others of the same type, so that they can be dealt with in batches of a size determined by considerations of economical production and of the requirements of production programming.

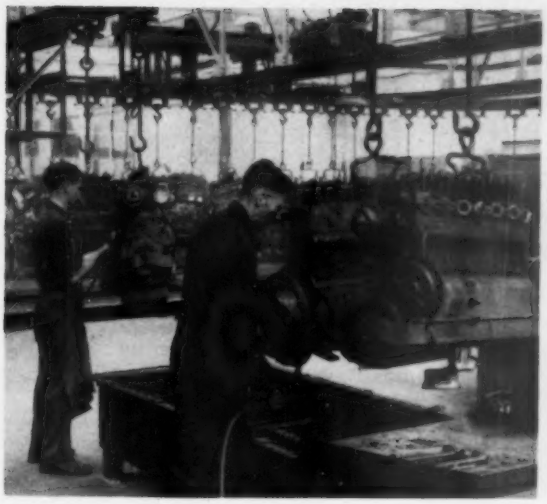
The sump, rocker cover, timing cover, side plates and core plugs are then removed to open up the engine. Usable covers and the sump are cleaned, inspected and, where necessary, manually reformed to shape and the flanges trued. They are then painted and subsequently passed into the stores for reissue.

Engines are drained of oil at the preliminary strip deck and are then passed to the initial wash section, alongside the area

Parts stripped from the incoming used engines are sorted and inspected before being passed to the rework section

Cylinder bores are machined, to either a standard oversize or for re-sleeving, on multi-spindle, production-type machines





In the preliminary stripping section, sumps, covers, and other items are removed, and the engine is then passed through the preliminary washing section

View of part of the crankshaft regrinding section. Here the standards of accuracy and tolerance maintained are the same as for current production shafts

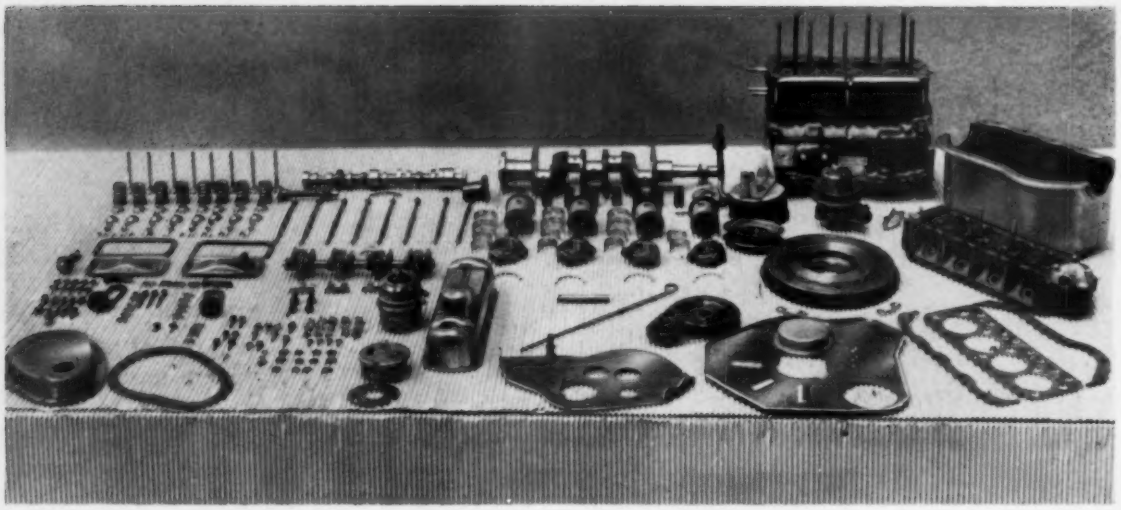
where the sheet metal parts are cleaned and reworked. There they traverse a washing booth in which they are sprayed with a hot alkaline solution. They emerge superficially clean and ready for dismantling. In the next section are six tracks on which the engines are stripped down to individual items.

All usuable components are collected and segregated in special containers for further treatment. It is routine practice to unreservedly discard a large number of working parts as no longer serviceable or worth reclamation. Examples of such components are all pistons, valves, and bearings. Typical of the auxiliary equipment is the water pump. Of this, only the main body is retained, all other parts are thrown out as scrap. All discarded parts are segregated as regards material. Some items are of relatively valuable alloys and care is taken to ensure that they are collected for disposal as scrap in the best market. This practice is an important factor contributing to the holding of the cost of replacement as low as possible.

Parts retrieved are then thoroughly cleaned by passing through a series of baths of appropriate cleaning solutions and solvents, arranged in three groups. In this section oil, sludge, rust, and carbon are removed. All parts are degreased in a bath of trichlorethylene followed by two baths of an alkaline solution at 175 deg F. Finally, they are conveyed through a Dawson washing machine—one machine for each of three groups of baths—and are delivered to the detail inspection section. Here each part is scrupulously examined and checked for suitability for further use.

Accepted parts are then passed to the appropriate reworking sections. These are laid out sequentially in one huge shop and employ 232 machine tools capable of working to consistently fine limits of accuracy. First are long parallel lines for cylinder heads, and alongside are further lines for

Finished replacement engines being loaded on to a lorry at one of the four dispatch decks



A percentage of the finished rebuilt units is stripped by the inspection department, and all parts are laid out for examination by control staff and management

cylinder blocks. Cylinders are rebored to standard oversize dimensions of +0·030 in or +0·060 in in the bores. Cylinders that have previously been bored to the maximum oversize are sleeved to restore the bores to standard diameter, and sleeved engines are rebored to +0·030 in oversize. Should an oversize, sleeved block be received it is re-sleeved to bring the bores back to standard again.

Following the cylinder head and cylinder block lines are lines for reworking camshafts, flywheels, and crankshafts. Practice as regards the regrinding of crankshafts differs between those of Austin and Morris manufacture. Austin shafts are reground to two standard undersize diameters, −0·020 in and 0·040 in but on Morris shafts the diameters are reduced in six standard steps. A superficial injury or surface indentation on a shaft may be rectified (in the metalling section) but it is held to be undesirable to build up shafts that have been reground to the maximum undersize. Consequently, no resort is had to that practice. Standards of accuracy and finish in the machining operations are precisely the same as for new engines in current production.

Next are lines for the smaller details and for the connecting rods, followed by other lines for rocker gear and for tappets. These occupy approximately half the width of the shop and the corresponding area accommodates the reclaim section, where minor repairs, such as removing broken studs and refixing loose studs are effected and the metalling section where welding and build-up operations are conducted.

Reworked components and the necessary new parts are progressed to the bulk stores and from there are issued as required to the marshalling stores where complete sets are made up for the large variety of engines covered by the replacement scheme. The baskets of marshalled parts are given a final wash by hot alkaline-solution sprays and then passed to the five assembly tracks. On these the engines are rebuilt, using similar methods to those employed on the assembly lines for new engines.

Directly from the end of each assembly line, the engines are passed to a series of beds for a running-in period; flushed, drained, and charged with oil in a so-called "sump section"; and then to beds for a production test to ascertain that power output and performance conform to required values. A percentage of the engines is regularly taken out for full-scale power tests to ensure that production standards are being fully maintained. Any unit failing to attain the requisite performance on the production test is withdrawn and passed to the rectification section alongside for examination and adjustment and is then again submitted for test.

On satisfactory completion of the production test, engines are passed to three painting lines where they are spray-painted overall. From the paint lines the units are placed on a transverse track and fed on to longitudinally arranged, four-tier storage racks. As required, engines are drawn from the opposite ends of these racks, transferred by overhead hoists to the four immediately adjacent dispatch decks, and loaded directly on to lorries for distribution by road. Alongside the storage racks is a packing area where engines are crated, or boxed for delivery overseas.

Since the rebuilt engines carry the same guarantee as new engines, control of operations is strictly maintained. As a regular routine, finished engines of each type are selected at random from storage and are completely stripped down, some by the inspection department and some independently by the engine research department. All component items are laid out for examination by control staff and management.

The treatment of gearboxes returned for reworking is similar to that for engines. However, gear wheels and pinions can only rarely be re-used, and replacement gearboxes, therefore, contain a higher proportion of new parts.

In itself, the manufacture of new, non-current parts for replacement engines and gearboxes is a huge task. These same parts are also required by B.M.C. Service Ltd. for its world-wide service parts organization. Non-current components are produced at the Dunbar Avenue factory.

TORSION BARS

AN interesting booklet entitled "Torsion Bars" has recently been published by The English Steel Corporation Ltd., River Don Works, Sheffield 9. After a brief introduction dealing with basic types, and some of the variants of torsion bar design, as well as with manufacture, inspection and corrosion protection, the book goes on to give some extremely useful design data. These data include basic design formulae, some notes on fillet correction, pre-setting, alignment of splines after pre-setting, permissible stresses and some general design considerations. The work concludes with some tables for conversion of metric units to English and *vice versa*, the British Standard Specification for Coarse Serrations, B.S. 2059, and the German Standard Specification for Serrated Hole and Serrated Shaft profiles.

CURRENT PATENTS

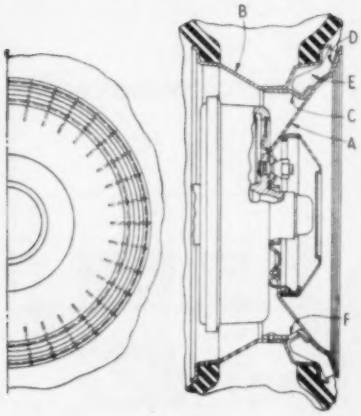
A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Ventilated wheel and rim assembly

By mounting the rim inboard and spaced from the wheel body, external appearance is enhanced and a flow of air is promoted over the brake drum. Wheel body A is a flanged and flared cone having a maximum diameter approximately equivalent to that of the tyre-bead retaining flange of the drop-centre rim B. This is supported and spaced from the body by an annular member C. It is suggested that member C is secured to the body by welding and rim B to member C by means of rivets.

Secured to the inboard face of the peripheral margin of the body is an annular member D, having circumferentially spaced, radially extending vanes E. Communication between the interior of the assembly and the space between rim and body is by way of apertures F in member C.

A hub cap retained by spring clips is



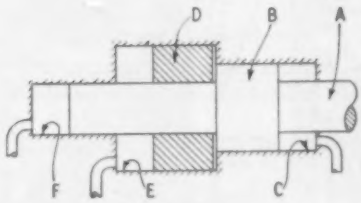
No. 791141

fitted over the wheel centre, and the outboard face of the body is finished to present the appearance of a cover plate. *Patent No.* 791141. *Kelsey-Hayes Company*, (U.S.A).

Hydraulic actuation of gear selector mechanism

A variable-speed gearbox having axially slidable selector rods each carrying a selector fork for effecting changes of gear ratio, has hydraulic, or pneumatic, actuation controlled remotely by a valve and a change-speed lever making an arc by step-by-step movements up or down through the series of gears. Each selector rod A is provided, at a suitable distance from one end, with an enlarged, cylindrical portion B slidable in a housing bore C. Between portion B and the end of the rod is an annular spacer D making a fluid-tight sliding fit internally on the rod and externally in a housing bore E. The extreme end of the rod slides in a bore F. All three bores are coaxial with the rod and each is furnished with a conduit for the supply and exhaust of pressure fluid.

The diagram shows the rod in neutral position. Admission of fluid to bore C moves the rod to the left-hand engaged position, carrying the spacer D with it,



No. 791008

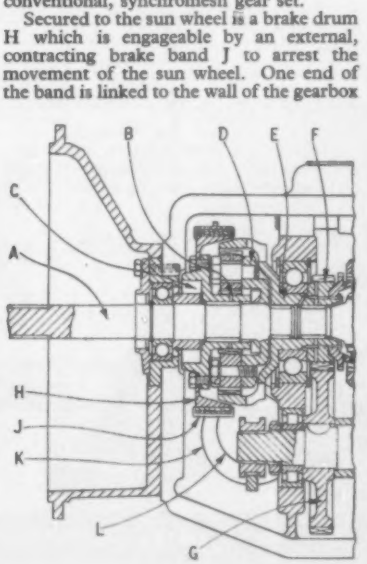
while admission of fluid to bore F moves the rod to the right-hand engaged position. Neutral position is obtained by admitting pressure fluid simultaneously to bores C and E, when the pressure differential arising from the different diameters of cylinder B and spacer D causes the spacer to assume the position shown and serve as an abutment defining the neutral position of the rod.

Although not mentioned in the specification, the end of the bore E adjacent to bore C will, presumably, be vented. This would appear to be necessary to obviate the possibility of either a positive or a negative pressure arising which might delay or prevent full movement of the rod. *Patent No.* 791008. *David Brown and Sons (Huddersfield) Ltd.*

Overdrive mechanism

Totally enclosed within the gear casing, this planetary-type overdrive gearing is arranged on the input end and thus provides twice the normal number of forward and reverse gear ratios. The input shaft A carries the sun wheel B on a uni-directional roller-type clutch C, the planet carrier D, and the ring gear on a freely rotatable tubular shaft E. This member constitutes the output shaft of the overdrive gear and carries the primary pinion F meshing with the layshaft constant-mesh gear G of a conventional, synchromesh gear set.

Secured to the sun wheel is a brake drum H which is engageable by an external, contracting brake band J to arrest the movement of the sun wheel. One end of the band is linked to the wall of the gearbox



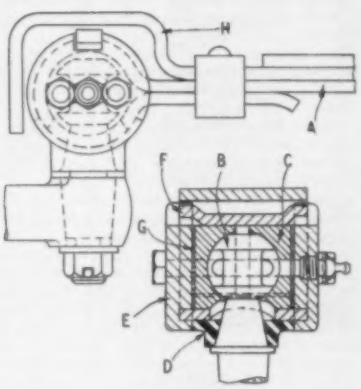
No. 791358

while the other end has a linked strut engaging the underside of a spring-returned piston in a hydraulic cylinder K mounted in a side wall of the gear casing.

The hydraulic cylinder is included in the circuit of a pump L driven by an eccentric mounted on the forward end of the layshaft. A manually operable valve, interposed between the pump and the cylinder, enables the brake to be applied at will in order to engage the overdrive. Alternatively, operation of the brake may be effected pneumatically, electrically, or mechanically. *Patent No.* 791358. *Moss Gear Co. Ltd.*

Independently sprung front wheel mounting

In this independent front suspension system, the respective halves of a transverse, laminated spring, anchored centrally

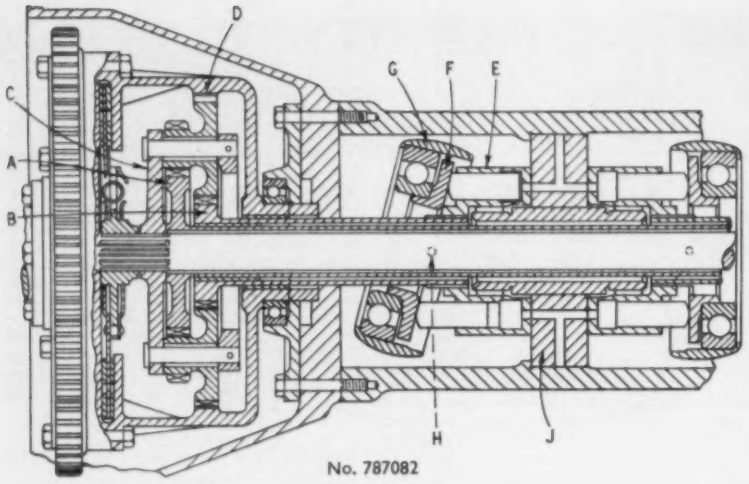


No. 791085

to the vehicle framing, constitute the upper members of the parallel-motion type control arms. The wheel carrier is provided at each end with a ball-joint assembly, each ball end being centred approximately on the longitudinal axis of the associated control arm.

At the extremities of the transverse spring, the main leaf A is completely looped to form an eye, which on its lower side is apertured to admit the ball stud and afford clearance for the requisite suspension movements. The ball end B is located in a complementary pair of concave seatings C inserted in the spring eye from opposite ends. A rubber sealing member D is fitted over the ball stud, and the assembly is completed by a pair of clamp plates E. These are grooved to receive the ends of the spring eye and are located by peripheral slots engaged on tongues F upset from the spring eye. Shim discs G, between ball seatings and clamp plates, provide a means of adjustment.

Two clamp bolts, passing through plates E, shims G and seatings C, and skirting on each side the waisted ball end, secure the assembly. Centrally disposed on one clamp plate is a grease nipple. Against the contingency of a fractured main leaf A, an adjoining leaf H is partially wrapped around the spring eye as a safeguard. *Patent No.* 791085. *Morris Motors Ltd.*



No. 787082

Hydro-mechanical transmission

An infinitely variable speed ratio, and corresponding variable torque ratio, over a wide range is obtained with this transmission comprising a hydrostatic drive unit combined with an epicyclic-type mechanical change-speed gear and with a friction clutch between input and output shafts. The epicyclic gear has a single compound train consisting of separate sun wheels A and B meshing with planetary pinions on a carrier C splined to the output shaft. An internal ring gear D, formed in the hollow input member bolted to the flywheel, is in engagement with the larger of the planetary pinions.

In the hydrostatic drive unit, two cylindrical drums E are each provided with an evenly pitched series of parallel bores in which plungers are mounted to slide. The plungers of each drum co-operate with a plate F mounted in a ball bearing in a carrier G which is inclinable about a fixed pivot H. Hydraulic connection between the bores of the two drums E is established by transfer passages drilled in a distribution block J secured in the casing in fixed relation to the pivots H.

Sun wheels A and B are rigidly connected by concentric tubular shafts to the rear and the forward drums E respectively. The drums are mounted on needle roller bearings on a sleeve member secured in distribution block J. Splined to the end of the output shaft, forward of the planet carrier C, is the driven plate of the hydraulically operated friction clutch. Engagement of the clutch will give a straight-through, mechanical drive.

Variable speed ratios between input member and output shaft are obtained by varying the angular dispositions of plates F. Except when either of these plates is normal to the axis of the drums E, or when the clutch is engaged, engine torque is transmitted partly by the epicyclic gearing and partly by hydraulic pressure between the opposed plungers of the two drums, one of which acts as a pump and one as a hydraulic motor.

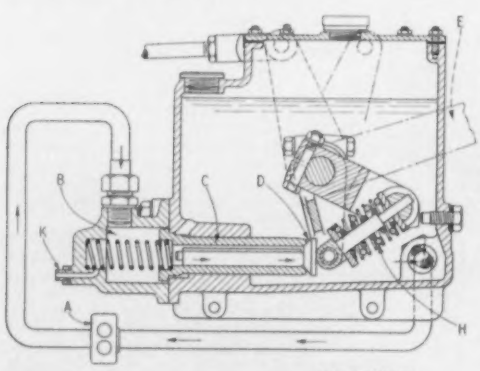
In either of two positions in which one plate F is normal to the axis, as shown, no pumping action or transfer of hydraulic fluid occurs and, providing the clutch remains disengaged, engine torque is transmitted wholly through the epicyclic gearing. These two positions normally will not correspond to limiting values but will be so located within the speed ratio range that the average transmission of power through the hydraulic unit will be a minimum over the full range.

The third position corresponds to the normal 1 : 1 ratio condition. By engagement of the clutch the hydraulic unit is unloaded and mechanically locked-up and maximum efficiency is obtained by a direct mechanical drive. *Patent No.* 787082. *Austin Motor Co. Ltd.*

Brake hydraulic servo mechanism

In the event of this hydraulic servo mechanism failing to function, an emergency mechanical brake linkage is automatically brought into operation. The hydraulic fluid is circulated by a pump A, which draws from the reservoir and delivers to cylinder B. In this cylinder is a spring-loaded hollow piston C, furnished with a reaction valve D. Actuation of the brake lever E, secured to spindle F, operates lever G to close, or partially close, valve D and apply braking pressure. Actually, the pressure is cushioned by springs H mounted between the end of lever G and parallel-lever member J, which is pivoted freely on spindle E. This spring linkage is intended to give the operator the desired 'feel'.

When the pedal is operated, valve D is closed, pressure builds up in cylinder B, and the vehicle brakes are applied by pressure fluid by way of delivery connection K. The reverse side of the piston head is open to the fluid in the reservoir. During a normal application of the brakes, the piston is held in a forward position, with its valve closed, against the pressure in cylinder B. Should the hydraulic circuit fail, or the pump cease delivery, pressure will be lost and the piston will be moved forward until a laterally-protruding lug L on member J engages lever M clamped and cottered to the sleeve of the emergency brake lever N. *Patent No.* 791257. *Fodens Ltd., E. Twemlow, and J. B. Mills.*



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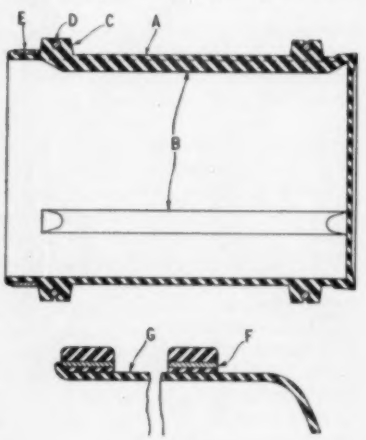

No. 792121

Rubber rumbling barrel

These rubber-bodied rumbling or grinding barrels are claimed to possess substantial advantages in comparison to the currently used metal barrels. They are not subject to corrosion, have a higher resistance to abrasion by the grinding media employed, are less noisy in operation, and are more easily cleaned.

The barrel, which may be of natural or synthetic rubber or a rubber-like material, comprises a cylindrical body A moulded from a flexible, vulcanizable rubber compound of approximately 90 deg Shore hardness, with three longitudinally extending integral ribs B on its internal surface. Moulded integrally on the outer surface are a pair of rubber tyres C adapted to contact the driving rollers of the barrelling machine. Each tyre is formed on its periphery with an annular recess for the reception of a rigid steel hoop D which serves to reinforce the tyre. A steel hoop E reinforces the open end of the barrel.

An alternative construction shows a steel hoop F sandwiched between the two parts of a tyre moulded at each end of a barrel G. *Patent No.* 792121. *Dunlop Rubber Co. Ltd.*

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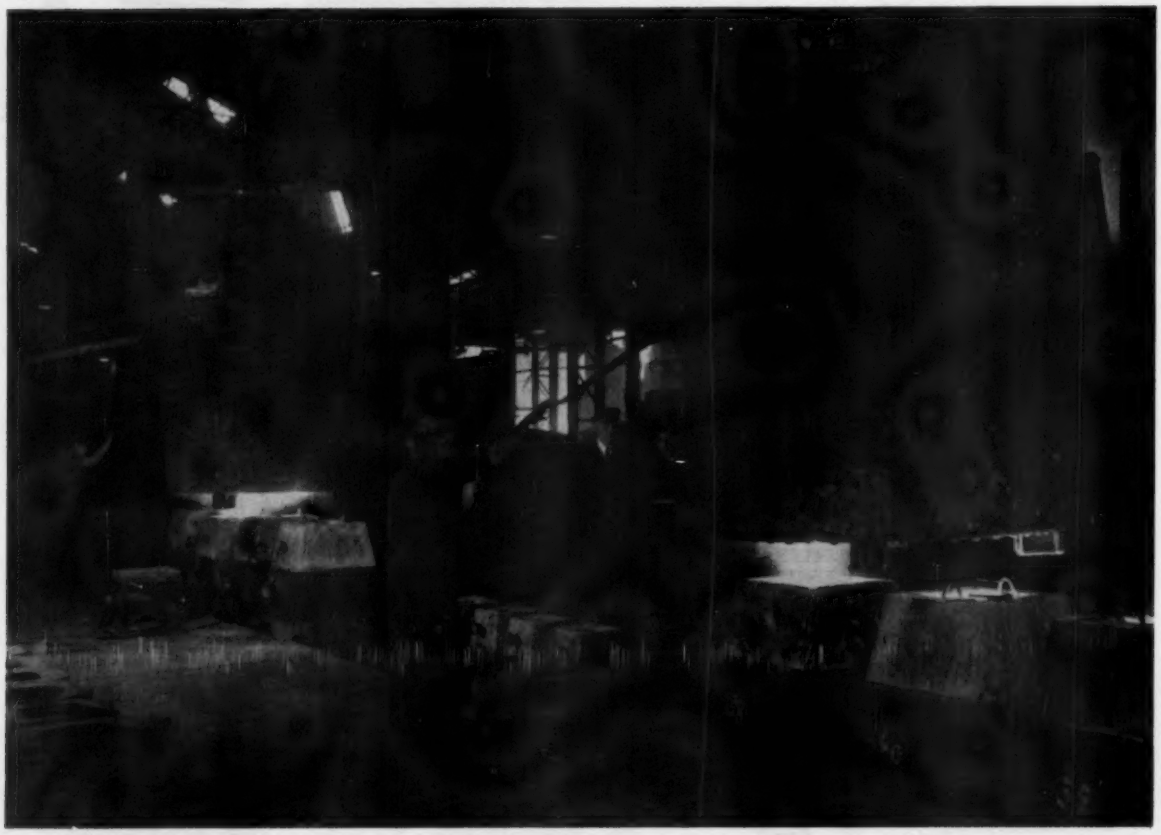

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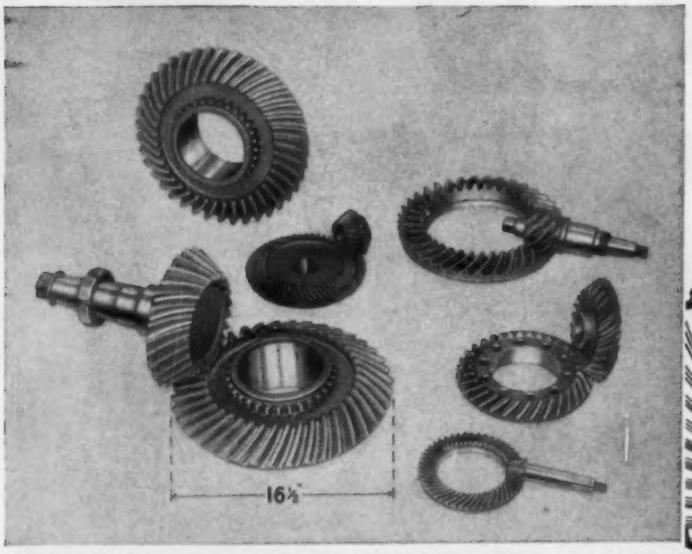

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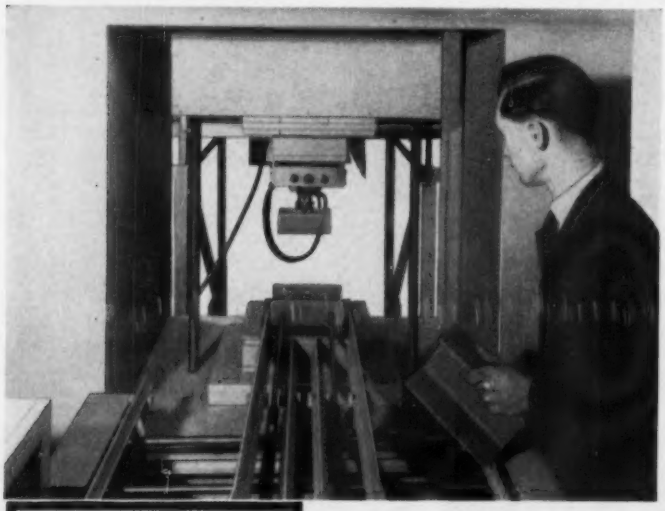

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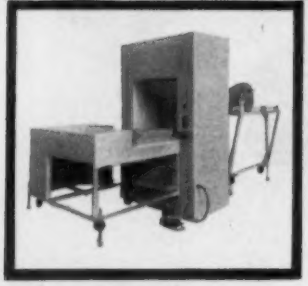

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The following is a translation of an Affidavit drawn up by Dr. Daniel Cespedes, Notario Publico, y de Hacienda :

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Lima, 9th April, 1957

PERUVIAN OMNIBUS COMPANY LIMITED
J. BLUME T.
Principal of the Transport Department

The undersigned, PUBLIC NOTARY AND COMMISSIONER FOR OATHS, CERTIFIES that Mr. JUAN BLUME TRAVERSO, proving his identity with his electoral papers number six hundred fourteen thousand five hundred and ninety four and his military papers number eight hundred and seventy seven thousand six hundred and sixty two, was present in the offices of the aforesaid Public Notary, with the object of having legalised his signature in the DECLARATION which he has signed as Technical Principal of the Transport Department of the Peruvian Omnibus Company Limited, ratifying before me the said declaration which he now repeats and says as follows : "Herewith we wish to certify that Omnibus number two, of "SCANIA VABIS" make, Motor number sixty two thousand one hundred and seventy nine, Chassis number ninety one thousand nine hundred and seventy, which operates on the Lima-Ica Line, has run one million and two hundred thousand kilometres without it being found necessary to dismount the engine for repair", which he signs again in my presence, in Lima, on the thirteenth day of the month of April of the year one thousand nine hundred and fifty seven.

J. BLUME T.

DR. DANIEL CESPEDES
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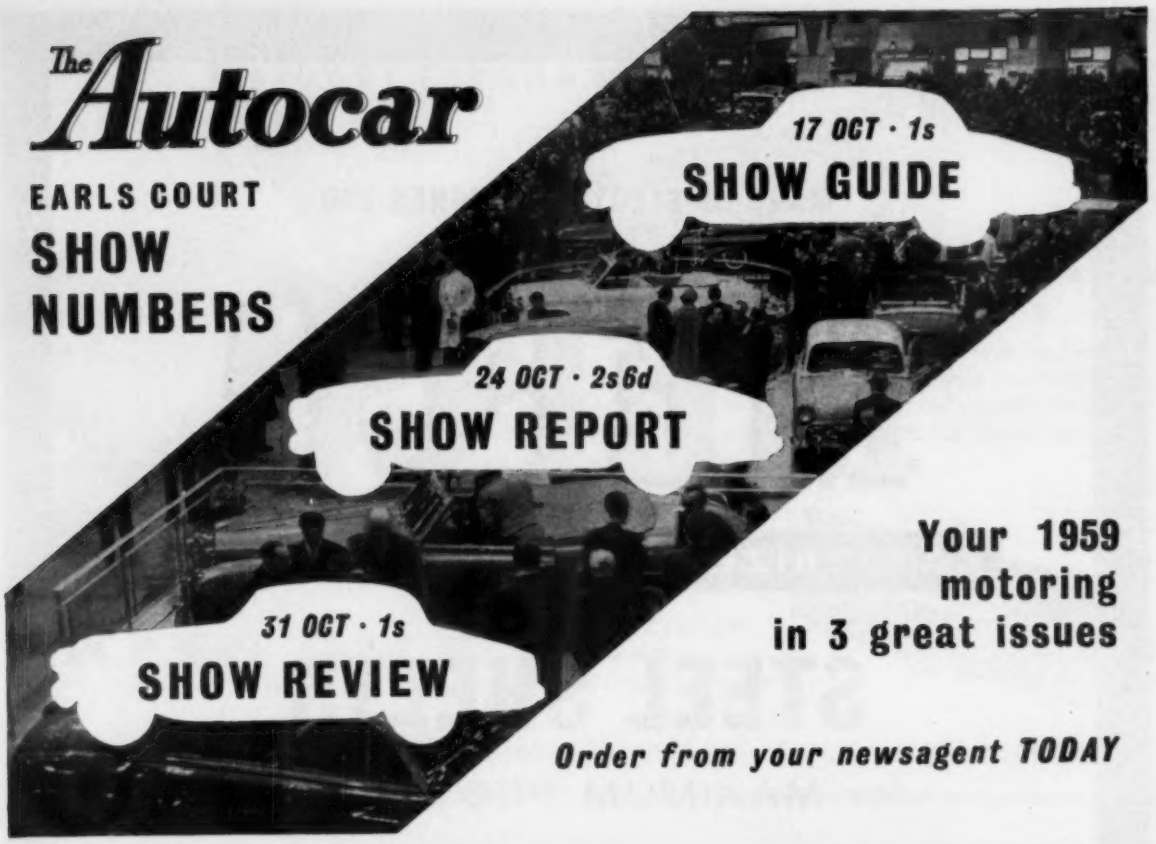
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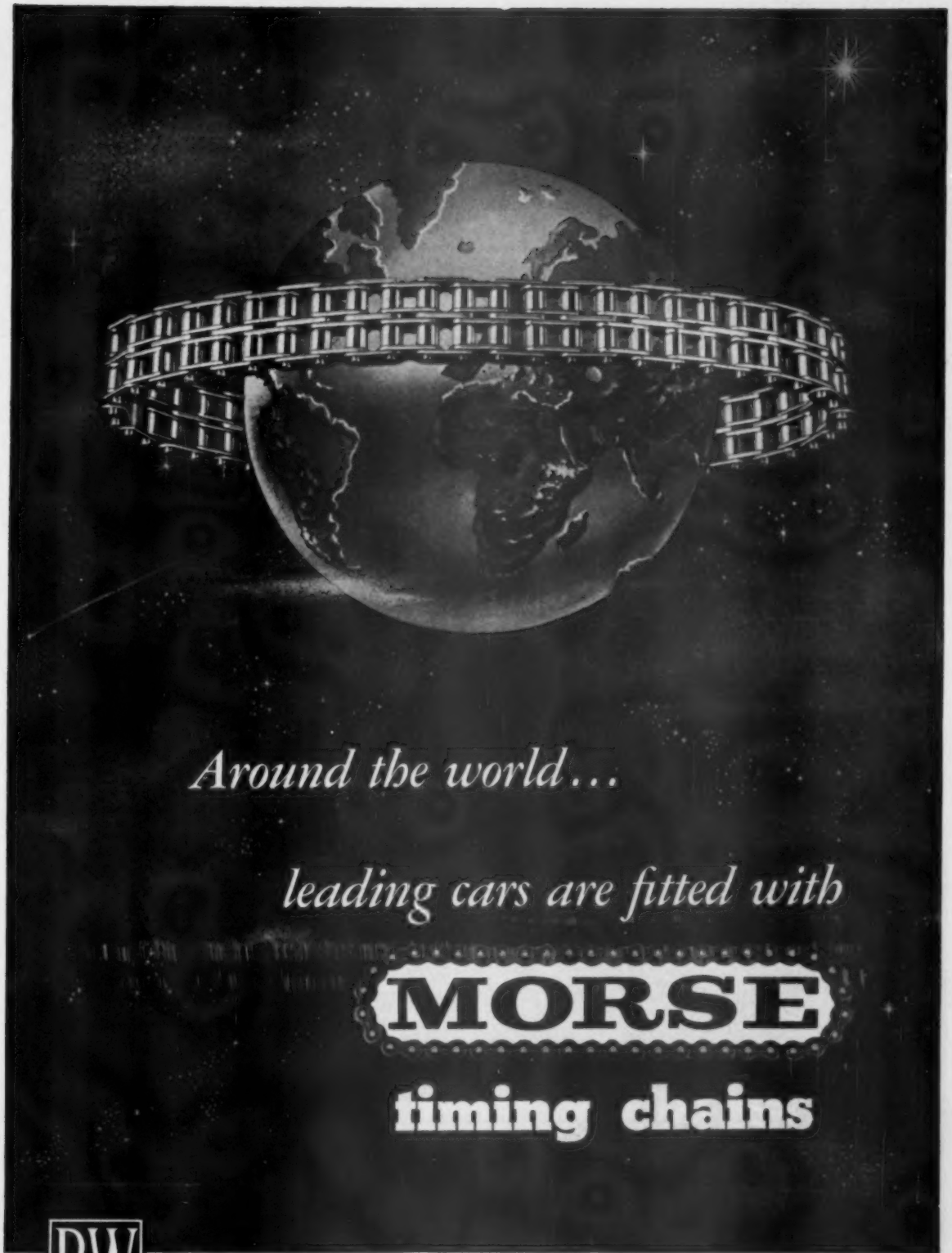

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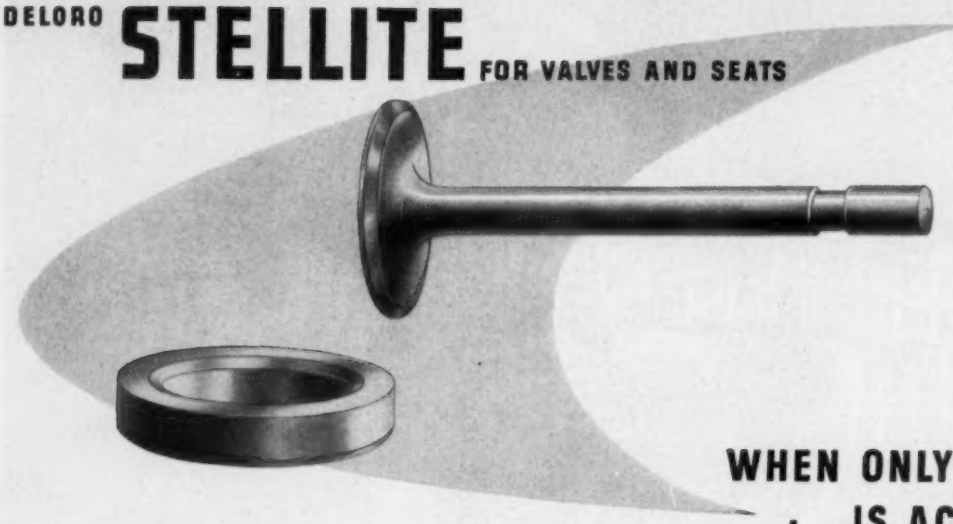

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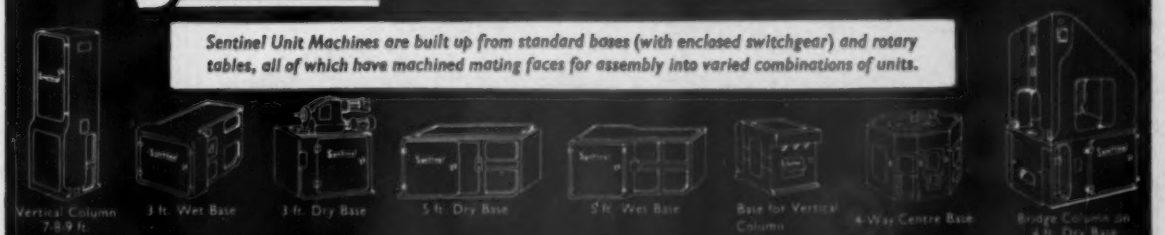
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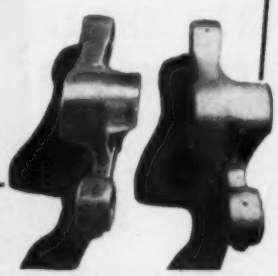

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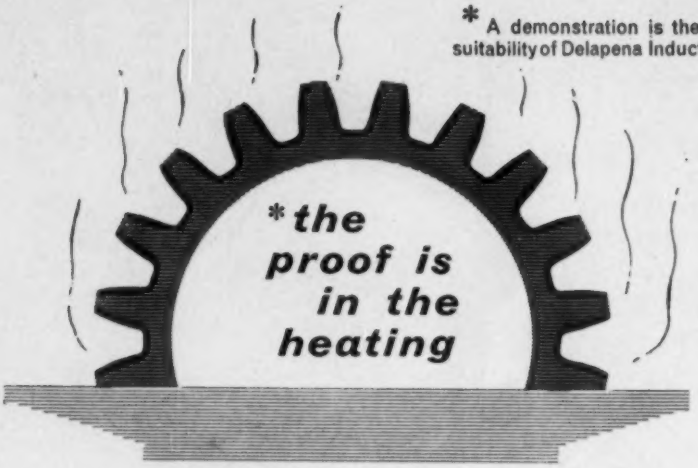

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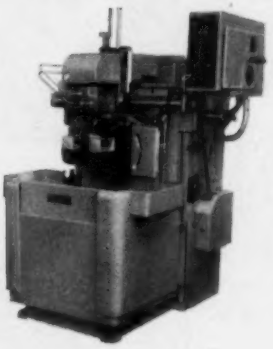
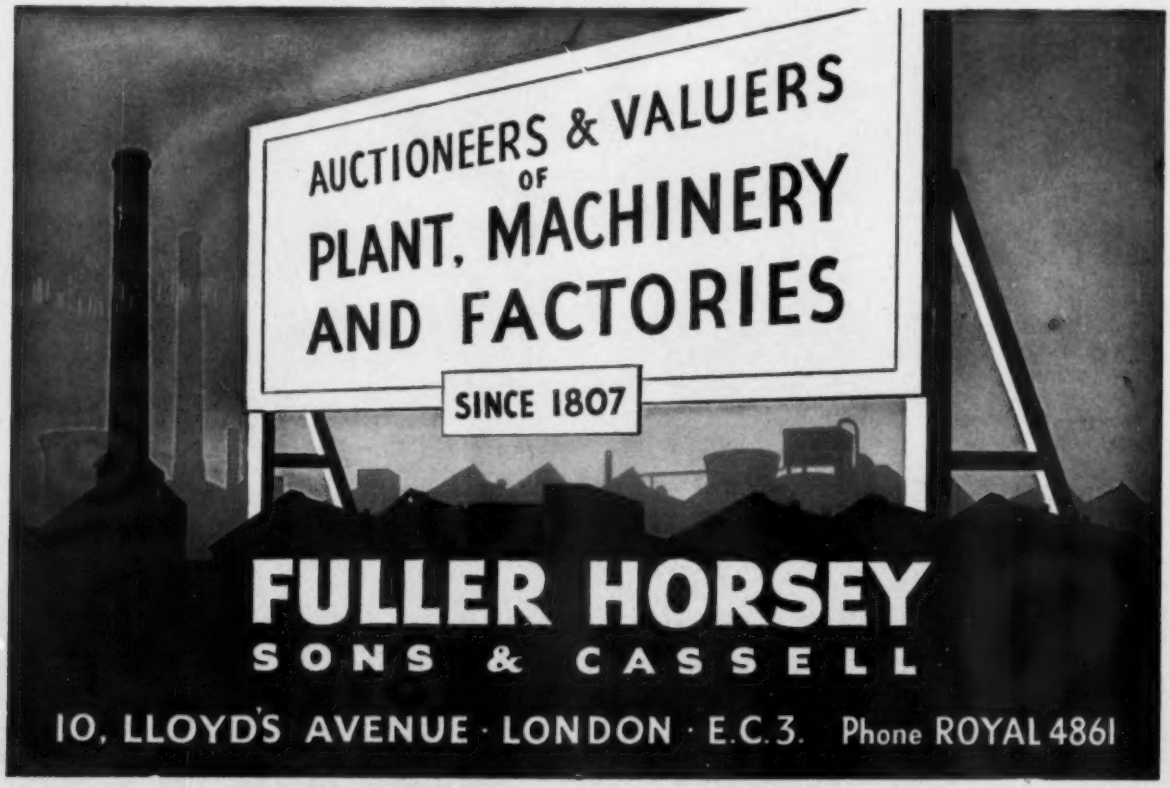


This fully automatic machine handles gears within a range of 1¼" to 26" diameter, 2 to 10 diametrical pitch, face widths up to 8", and spur and helical gears up to 30°.



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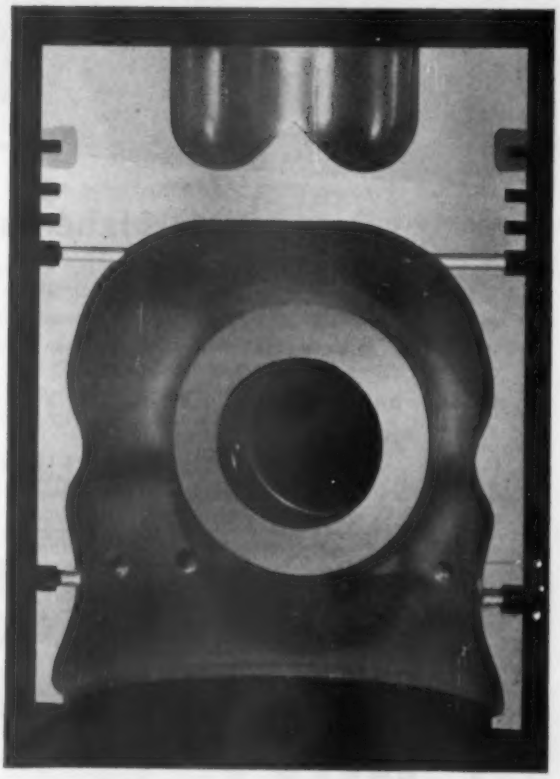

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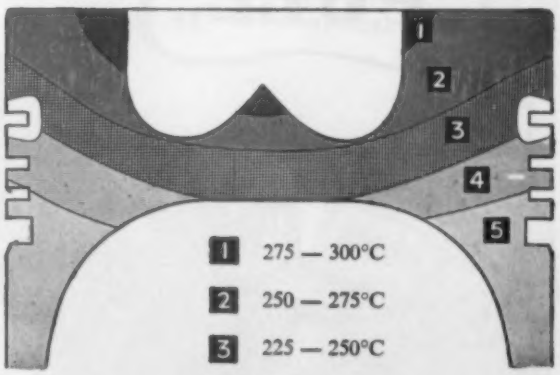

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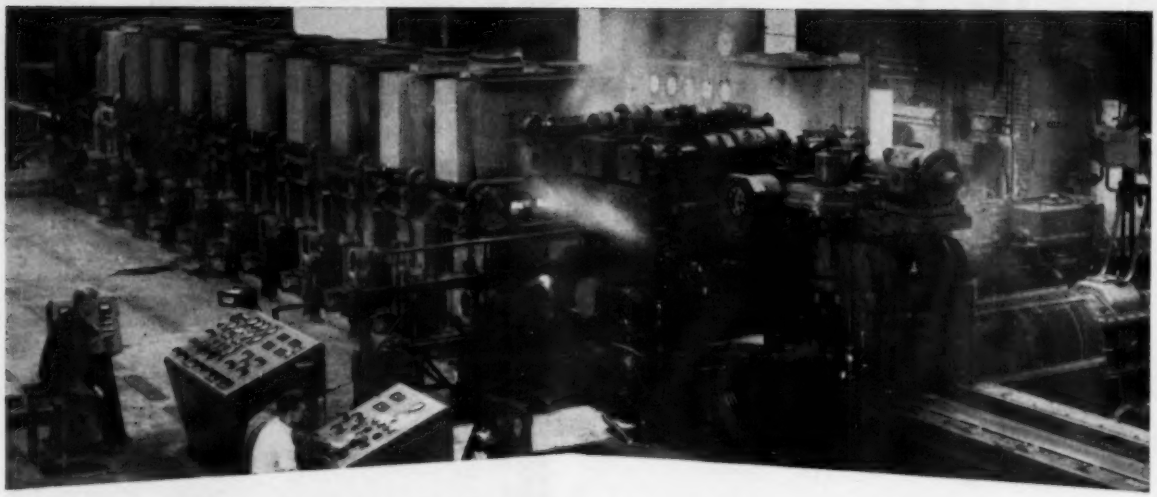

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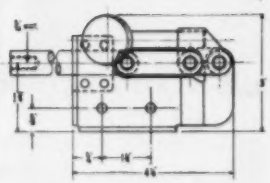
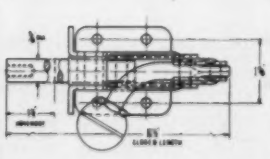


Closed position for "push" clamping. Neat and compact.



Plan view of above. Reverse position for "pull" clamping.

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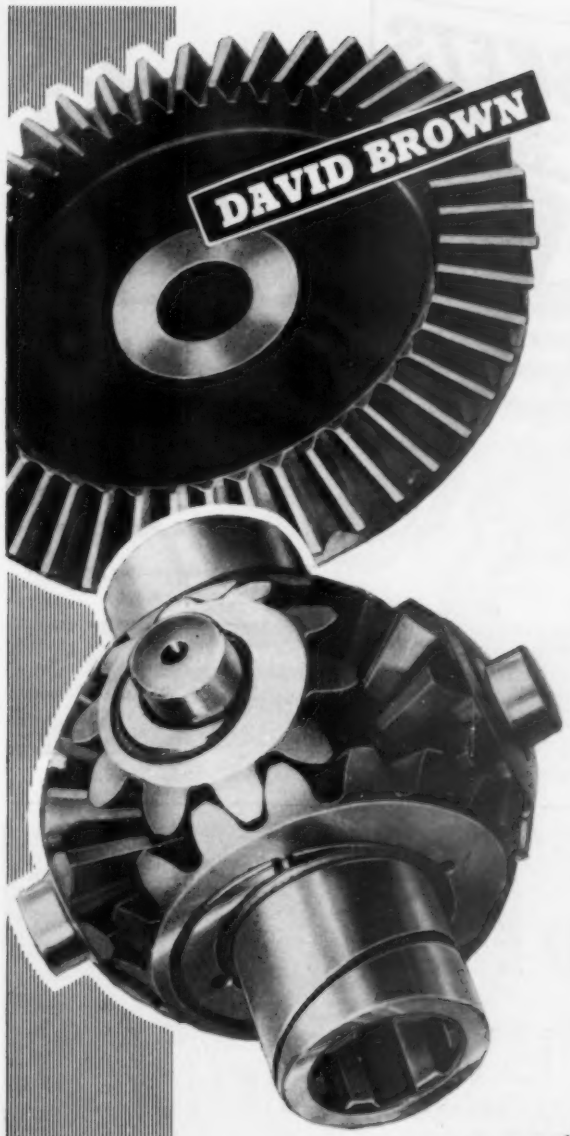

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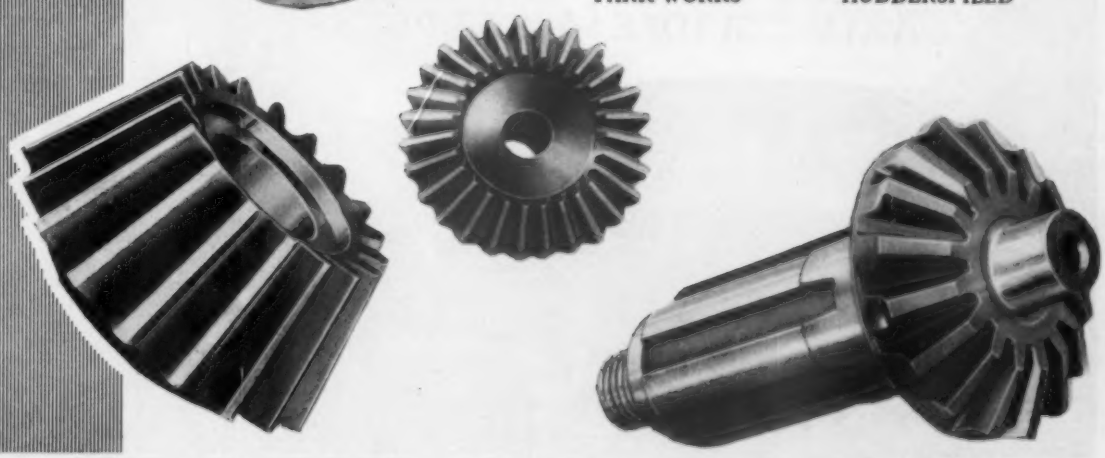

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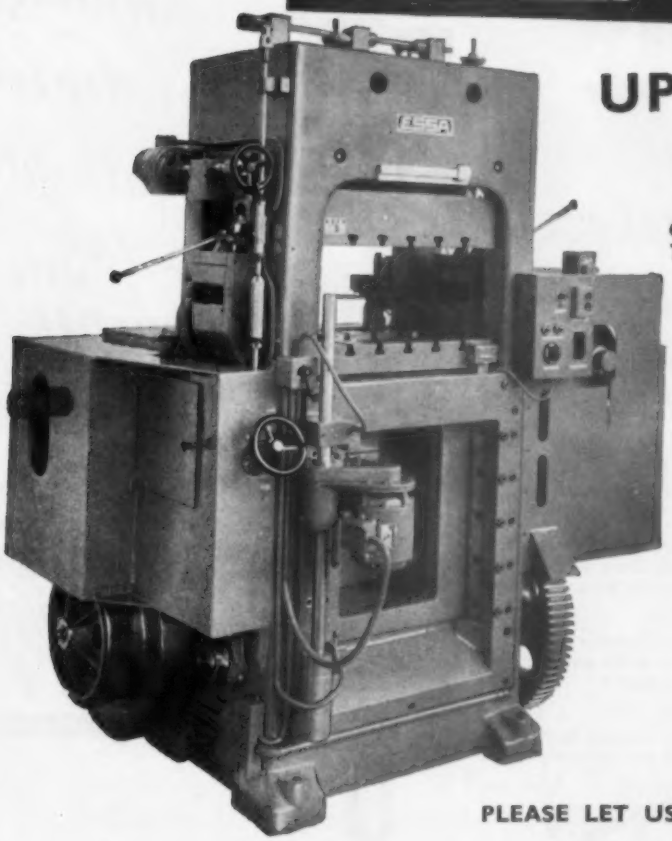

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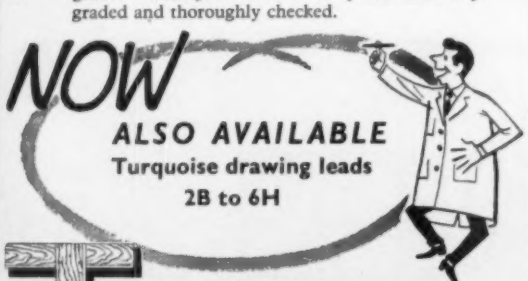

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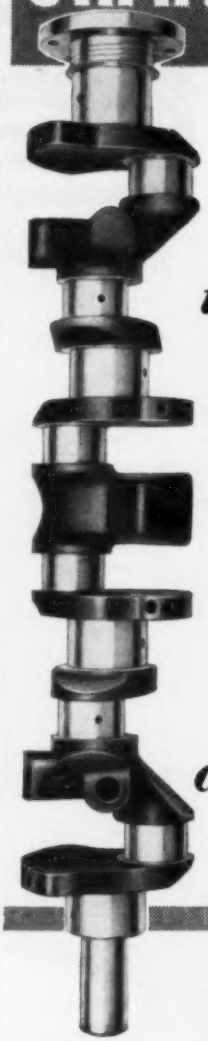

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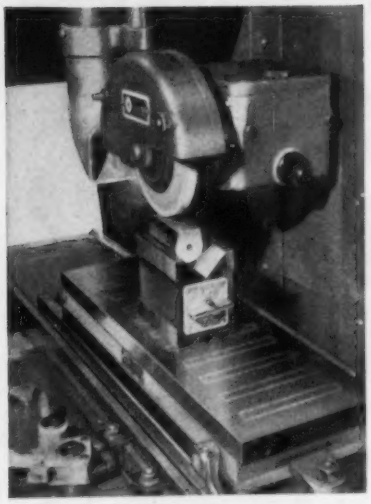

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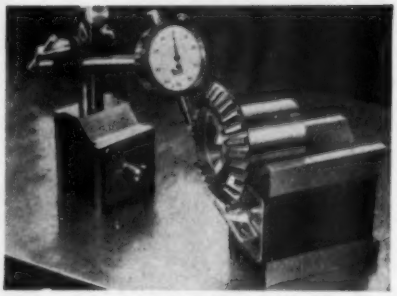

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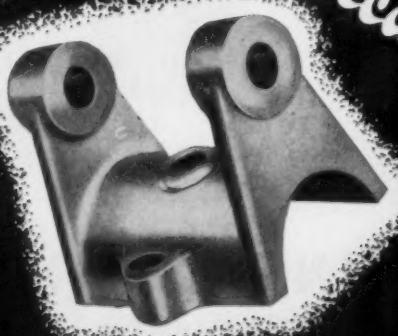

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'MG 5' MINDS THE STEPS OF THE 'DONINGTON'

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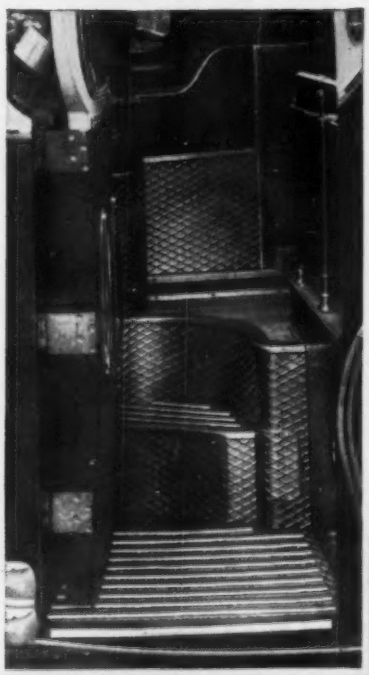

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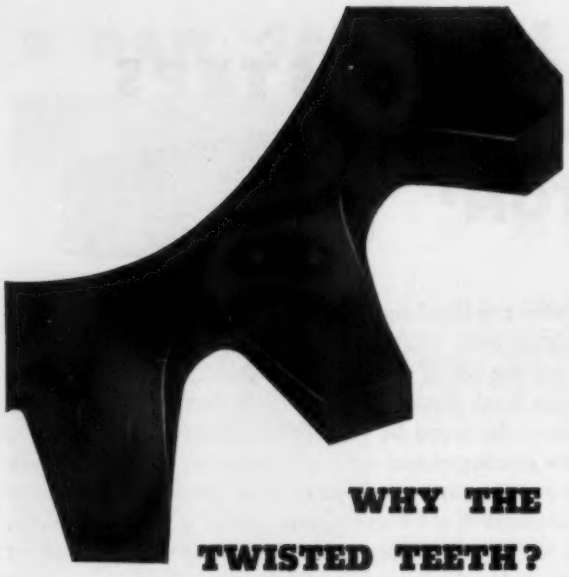

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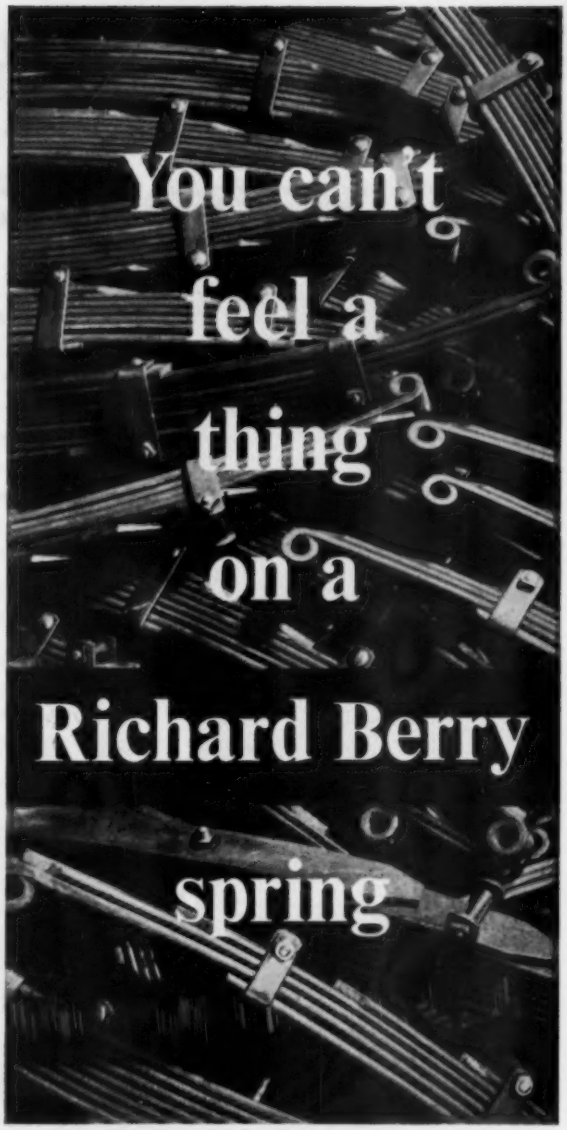

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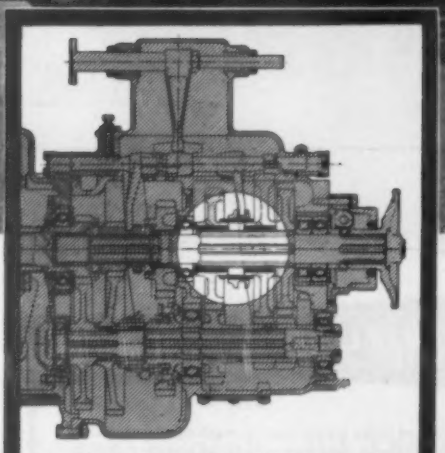

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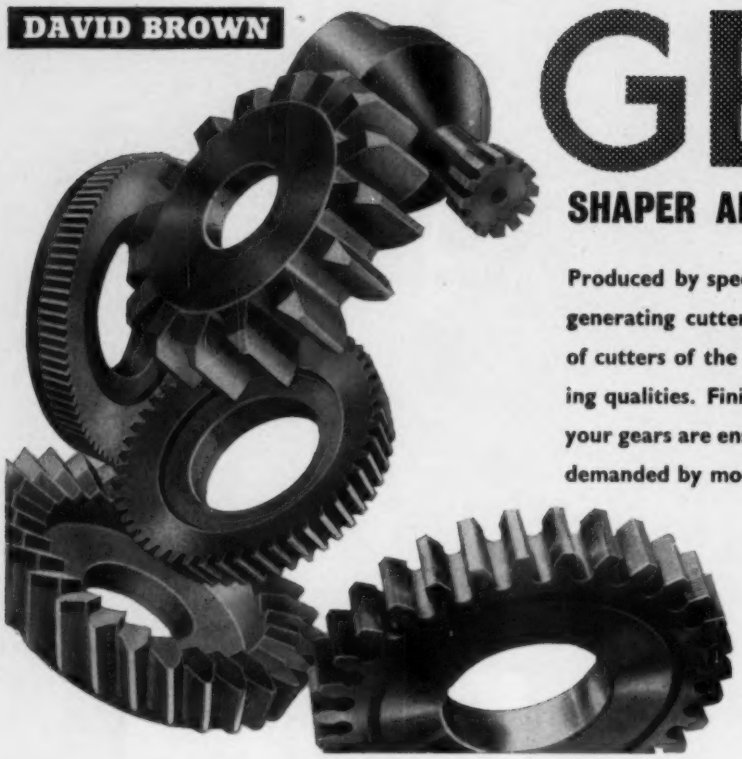

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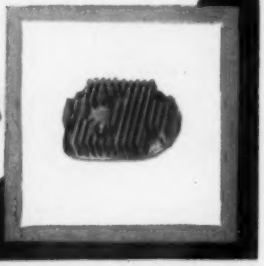

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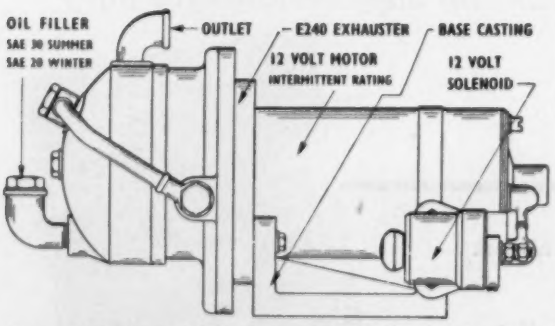

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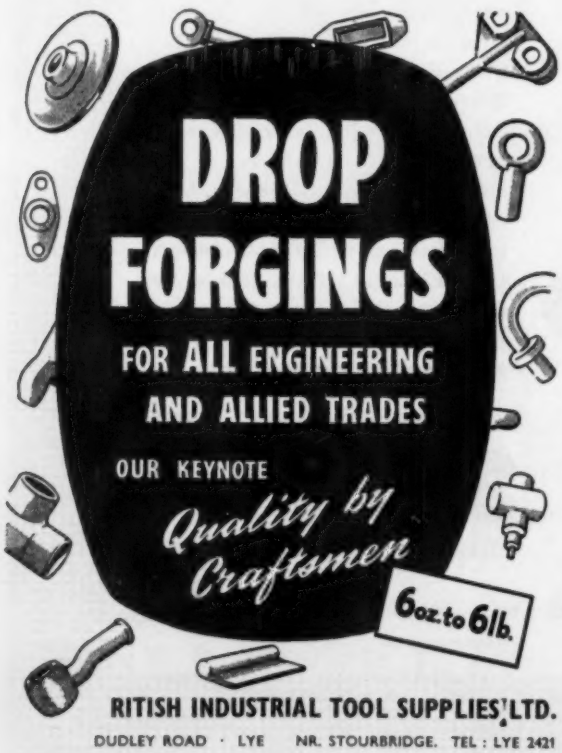

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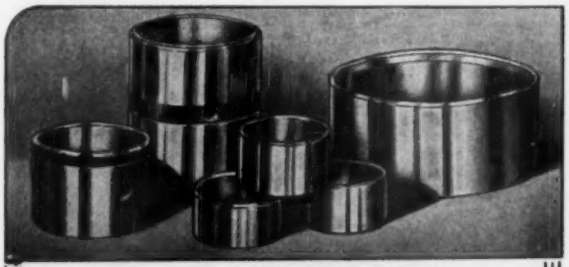

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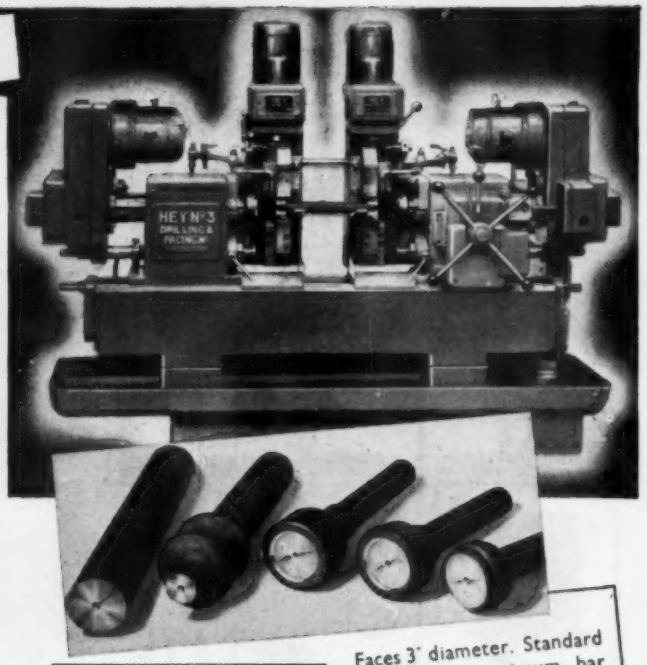

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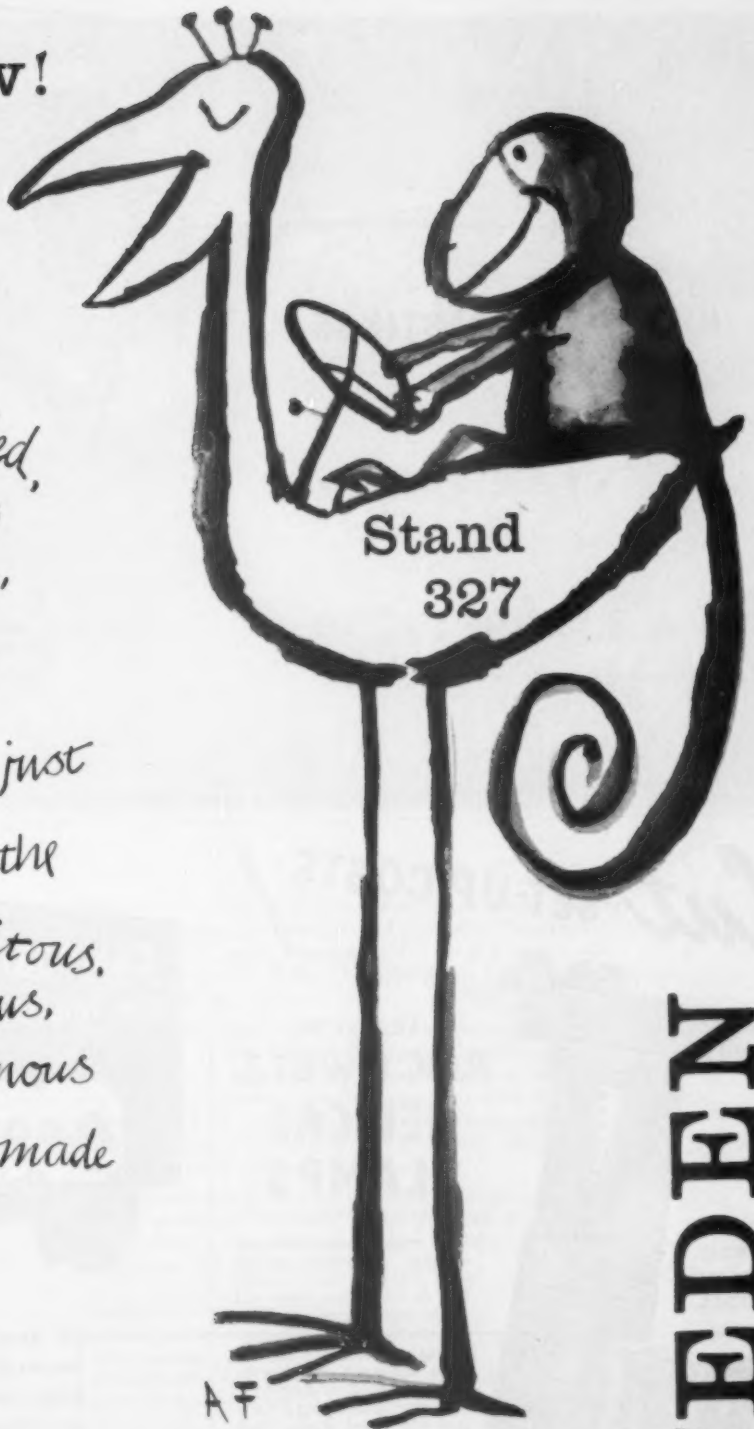

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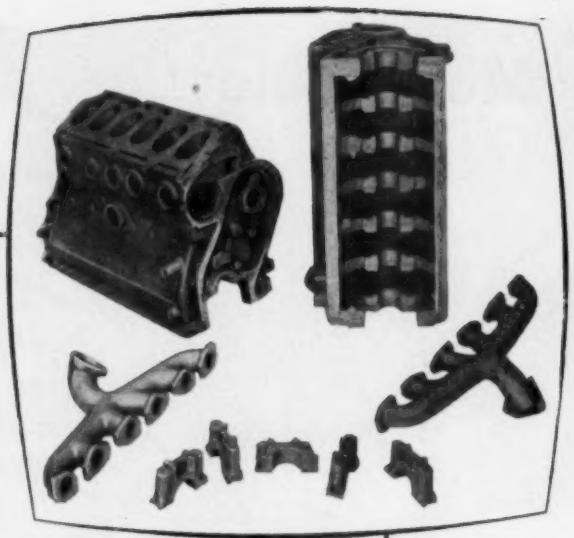

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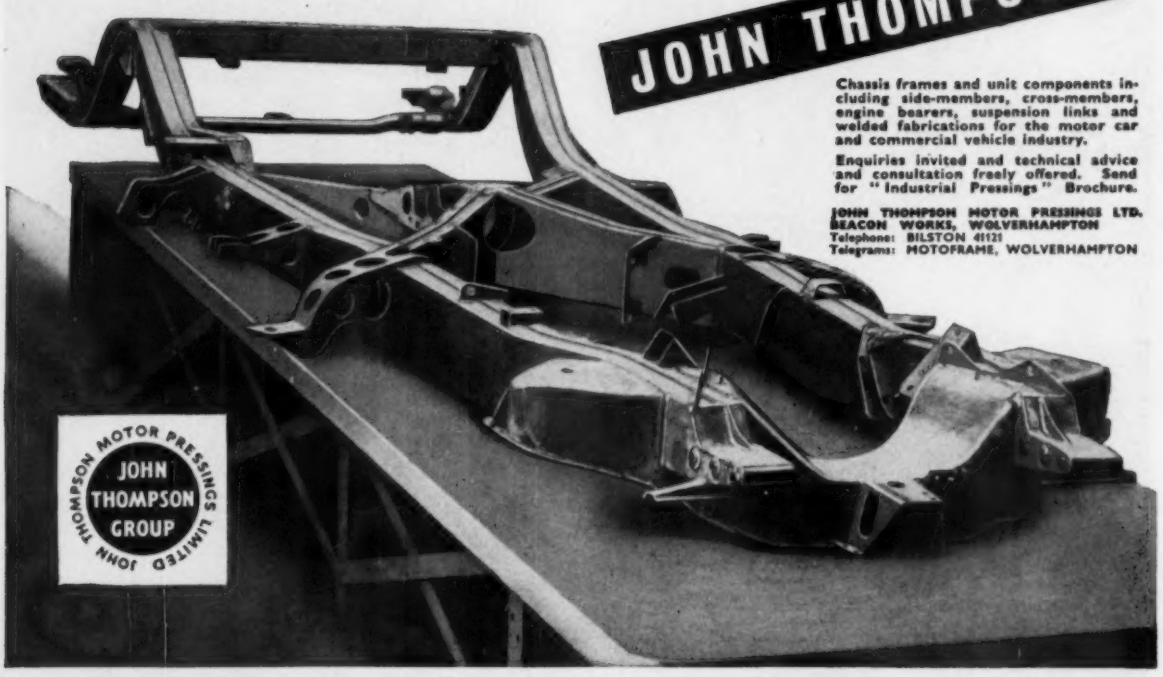

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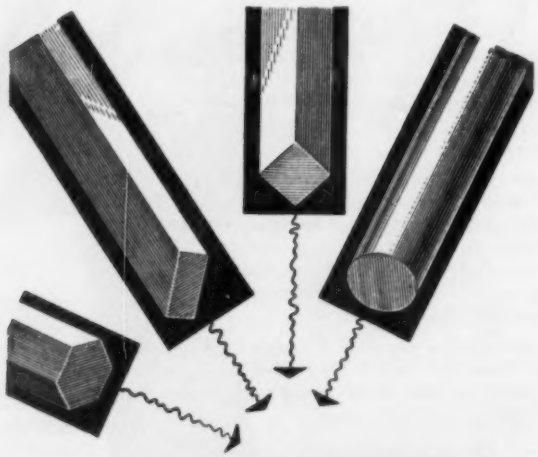

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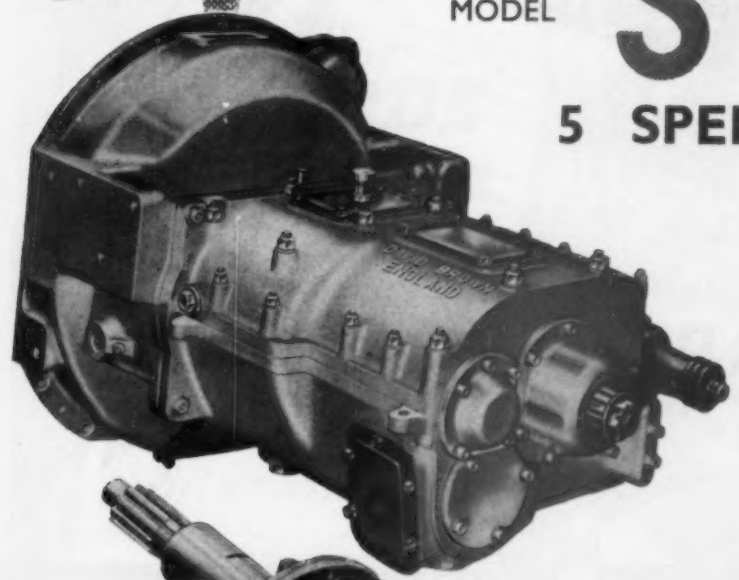

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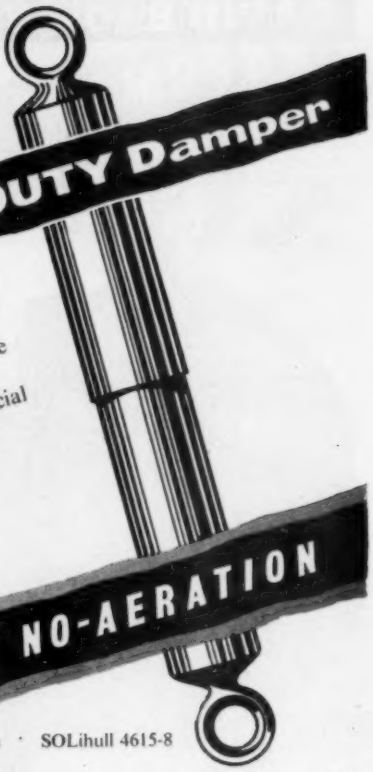

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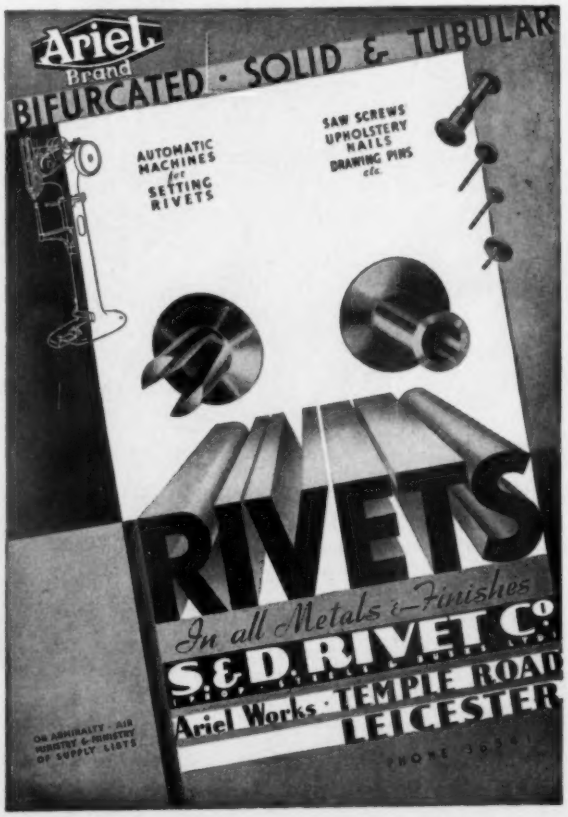

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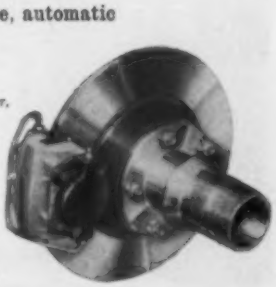

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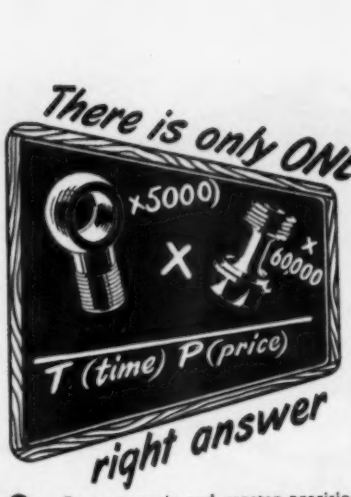

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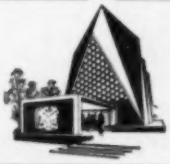

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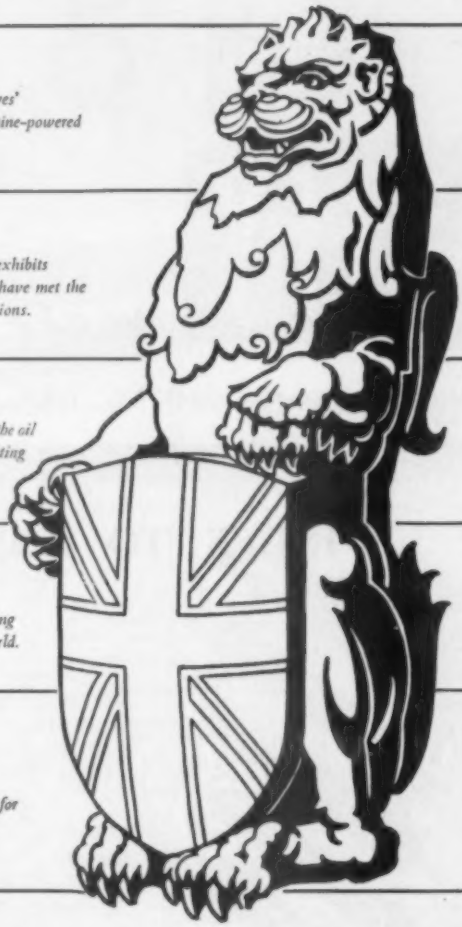

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This organisation aims to tell the real story of Britain's achievements, to claim for the British people and for British industry the credit that is theirs: and by doing so to reinforce Britain's prestige throughout the world to the benefit of our trade and our standard of living.

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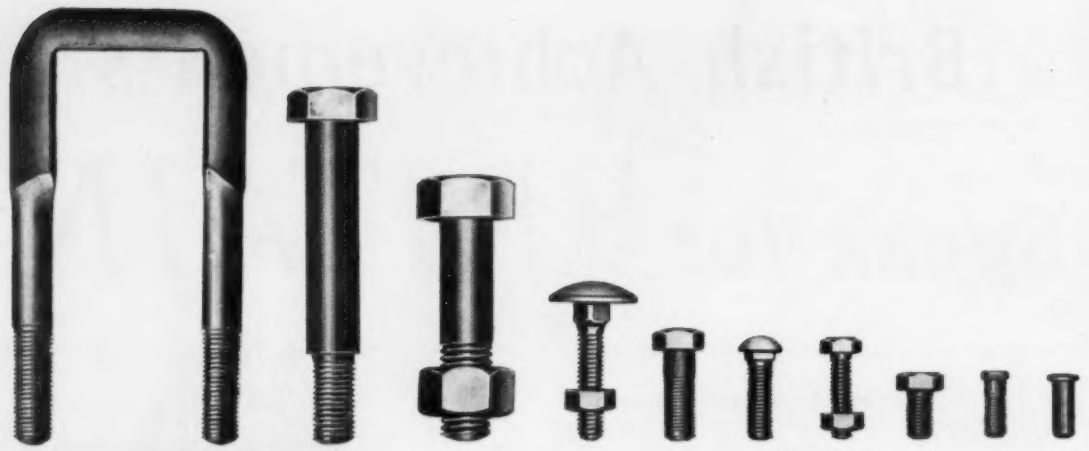

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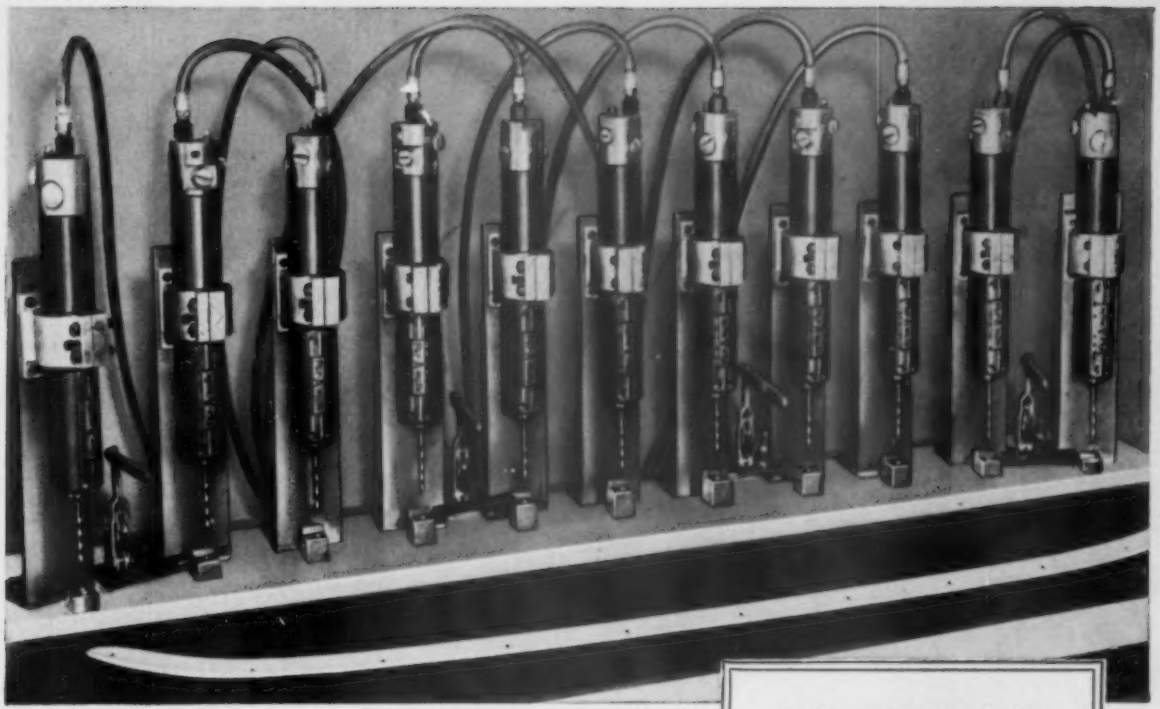

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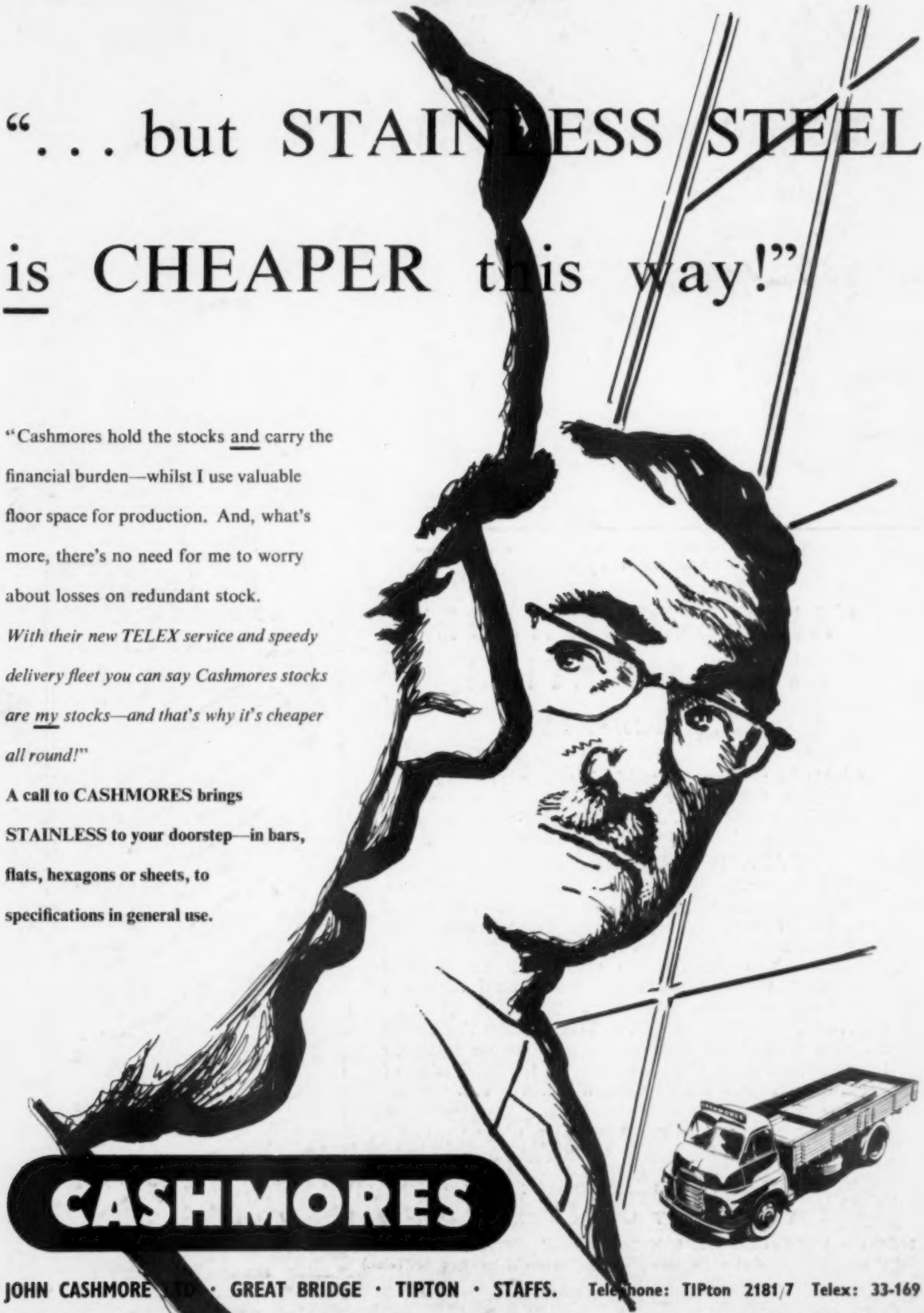

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